Heating Piping & Air Conditioning

with which is merged the Aerologist

Design, Installation,
Operation, and Maintenance:
Industrial, Commercial, Institutional,
and Public Buildings

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Published monthly by the Keeney Publishing Co., 6 North Michigan Avenue, Chicago 2, Illinois. Telephone: State 2-6916. Subscriptions: U.S. and possessions, $3.00 per year; Canada, Cuba, Mexico, Central America, South America, $4.00 per year; other, $6.00, Current copies, U.S. and possessions, 35c each; back copies, 60c each. January 1949 issue, $1.50 per copy. Member Audit Bureau of Circulations and Associated Business Publications, Inc.

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For enclosed radiators — Sylphon valves in angle and globe types are designated by Nos. 895 and 896. This valve is similar to the No. 885 in operating principle except that it employs two bulbs which are connected by flexible tubing with the trottling steam valve.

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THE EDITOR'S PAGES

JOHN HOWATT DIES

John Howatt—for the last 35 years chief engineer of the Chicago board of education—died Saturday, October 22, and services were held October 26 at Lake City, Minn., where he spent his boyhood.

In addition to his duties as chief engineer, Mr. Howatt served also in recent years as business manager of the board of education.

"The Chief" will be missed by many people in many ways. HPAC's readers will remember him as the author of several excellent articles on school heating and ventilation, as well as other subjects. He was a member of our board of consulting and contributing editors continuously from Vol. 1, No. 1.

BUILDING SHOWS SUBSTANTIAL GAIN

Spectacular increases in the dollar volume of building and construction contracts awarded in September in the 37 states east of the Rocky Mountains brought the total to $1,093,724,000, the highest reported for a single month since June 1942 when wartime construction was at its peak, according to F. W. Dodge Corp.

It was the second time in the Dodge statistical series dating back to pre-boom years in the 20's that more than a billion dollars in awards were reported for a single month in the 37 eastern states. The record volume cancelled deficits of earlier months to bring the cumulative dollar volume for the first three quarters of the year to 1 percent higher than in the corresponding nine months of 1948.

Investments by private owners accounted mainly for the advances, the overall contract total for projects classified as privately owned amounting to $804,970,000 to show a 37 percent jump over August and 60 percent increase over September of last year. Public account awards amounted to $288,754,000 or 9 percent less than in August and 11 percent more than in September last year.

Residential awards totaled $525,572,000 to reflect a 34 percent gain over August and 88 per cent over September of last year. Private residential building accounted for 96 percent of the month's volume. Substantial gains were reported for both apartment houses and single family dwellings.

Nonresidential awards totaled $345,023,000. This volume was 24 percent higher than in August and 23 percent greater than in September a year ago. Private nonresidential volume showed gains of 40 and 26 percent respectively over August and over September of last year while public volume was off 5 percent from August and up 17 percent over September of last year. Extraordinary gains were reported for commercial, educational, science and religious buildings.

Heavy engineering awards failed to follow the upward trend shown in residential and nonresidential building.

At the end of the third quarter of the year the cumulative total of awards was $7,410,708,000 in the 37 states against $7,345,773,000 in the corresponding nine months of last year, a gain this year of 1 percent. Public account awards were up 17 percent and private work was down 7 percent in the period. Nonresidential awards were running 5 percent less, residential awards 3 percent more, and heavy engineering awards 6 percent more than in the corresponding three quarters of last year.

IN WHICH WE SAY "THANKS"

Every year—not long after Labor Day—for about a decade-and-a-half we've dumped into the U. S. Mail hundreds of "listing sheets" addressed to manufacturers of heating, piping, and air conditioning equipment for industrial, commercial, institutional and public buildings. These sheets (they're really eight-page folders filled just about solid with not-too-large type) show by means of check marks the products of each manufacturer to whom they are sent.

Getting these sheets addressed and checked before mailing represents many man-hours—and woman-hours—of work in our own office. Many more hours of effort are chalked up by the manufacturers in going over the sheets to be sure the listings are accurate and complete. And, of course, when the sheets get back to us, there's still plenty to do before the final result—the Directory Section—appears in our January issue for your use throughout the year.

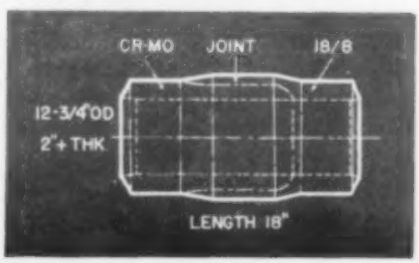
We bring this matter up here for two reasons. First, we want to remind you that a complete, accurate, easy-to-use, and up-to-date Directory Section of Heating, Piping, and Air Conditioning Equipment will appear again this January for the umpteenth time. Second, we want to thank the manufacturers for the tedious job they undertake in checking this lengthy listing sheet and advising us of their new products, etc.

While we're in the mood, we'd like also to thank the many readers of HPAC who recently answered our questionnaire about their use of this Directory Section. We were surprised ourselves to find how thoroughly and how often this editorial feature is referred to throughout the year it is current.

SPECIAL PIPE JOINT AT SEWAREN STATION

At the new Sewaren generating station, Sewaren, N. J., (described briefly on page 104 of the October HPAC), there are no valves between the boiler and the turbine stop valves in the main steam system. The pipe is made of 3 percent chromium—1 percent molybdenum alloy steel. The turbine stop valves

Transition from ferritic to austenitic steel pipe at turbine stop valve



THE EDITOR'S PAGES

are made of stainless steel which expands about 50 percent more than the ferritic steel of the pipe. The welded joint between the two was made in the shop by a new method, the purpose of which is to have the major portion of the interface between dissimilar metals longitudinal to the pipe rather than transverse, and therefore subject principally to shear stress as a result of internal pressure or bending forces.

The accompanying diagram shows the transition from ferritic to austenitic steel pipe at the turbine stop valve.

RE-USE WATER FROM A.C. PLANTS?

A public hearing before a committee of the Chicago city council last month on a resolution that had been introduced to permit the re-use of condensing water from air conditioning and refrigeration installations indicated little support for the idea and it is expected the matter will be dropped.

As in many cities throughout the country, however, the water required for condensing by large air conditioning installations presents a problem; in Chicago, overloading the sewers rather than a limited supply of water is the greater difficulty.

The sewer situation in Chicago is "impeding" the installation of air conditioning, said one consulting engineer in an informal statement at the hearing. He said that the use of cooling towers "is an excellent way out" of the problem although this tends to increase the cost of installation "by 5 or 10 percent." While he was in agreement with others who made statements that it would probably be unwise to re-use condensing water for drinking purposes, he felt that further consideration should be given to pumping the condensing water after its first use to an open tank in a building for further use for flushing toilets, cleaning floors, and for various process requirements in factories.

The subject of re-use of condensing water has received attention in several articles published in HPAC, and one article (by E. P. Heckel, in the December 1936 number) suggested that the utilization of *sewage* water as a condensing medium for air conditioning plants has possibilities. In an investigation along these lines, it was found that the temperature of sewage water was very close to the temperature of water available from the city mains. A proposed installation was, however, rejected for psychological reasons.

In an article in the April 1937 HPAC, E. Vernon Hill proposed taking condensing water from the city main, passing it through the condenser, and returning it to the main, the city to charge for the service on the basis of the amount of heat imparted to the water. A "kilotherm" meter would be used to measure the heat.

A short article in the September 1937 issue described an installation at the Oklahoma City city hall taking condensing water from the city main, using it, and putting it back into the main. Two pumps were installed to force the water through the condensers of this ammonia plant.

"QUOTE"

"It is difficult to see wherein construction market conditions in 1950 will be radically different from those which have prevailed this year.

"At this moment it looks as if 1950 will be in the main a year of further adjustment. There is reason to doubt whether the current upswing in activity can be viewed as the beginning of a sustained upward movement. Commodity prices, business activity and construction are likely to have mixed trends rather than trends that will be uniformly up or down. In general I would expect the year's volume of public building and engineering work to increase somewhat over 1949, the total of private construction to decline moderately. As a result, total contract volume may be moderately less, in dollars and in physical units, than this year's total volume.

"In the residential field, public housing projects will obviously increase and private housing activity is rather more likely to fall off moderately than to increase in any spectacular way. If rent control actually comes to an end there will be some adjustment of rental housing occupancy tending to lessen overall demand for rented space. At some stage the currently reduced rate of new family formations will result in a slackening of demand.

"What the continued buyers' market means to the investing public is, probably, a gradual lowering of construction costs. What it means for producers of materials is an increased urge toward improvement of products, new products, lowering of costs and, above all, progressive improvement in analysis of market potentials and of marketing methods.

"When I appraise next year as one of moderate change in construction market conditions, mixed trends and continuing readjustment, I imply that forces that will be at work in 1950 will prepare the way for some new development thereafter. Last year the American people invested only 8.3 percent of national income in new construction. In the peak years of the late 1920's they so invested over 14 percent. We are currently building to meet immediate and pressing needs; we are building very little to meet the needs of the future.

"By its high competence and its great flexibility the construction industry has demonstrated its capacity to produce all the new building and engineering structures demanded by the country's expanding needs. Construction's most vital job has always been provision of the physical facilities required by our dynamic, growing economy. Every improvement the industry can make in materials, marketing, design, construction methods and reduction of costs will contribute to the overall objective."—THOMAS S. HOLDEN, president, F. W. Dodge Corp., New York City, at the annual meeting of the Producers' Council.

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"OPEN FOR DISCUSSION"

USE OF PRESSURE WELDING IN PIPING WORK

A number of contractors are using pressure welding machines for making welded joints in radiant heating installations and in the piping for skating rinks. The process has the advantage of mechanizing a rather tedious and bothersome job of making numerous hand welds in small pipe. Like any mechanical process, the material to be processed must be prepared with a fair degree of accuracy and uniformity.

In the case of pressure welding the joint surfaces should be smooth, square with the axis of the pipe, and clean. If the joint surfaces are prepared some time before they are to be welded, a coating of grease should be applied to prevent rusting. This grease must be removed with a suitable solvent before welding the joint. A 6 or 7 deg included angle can be used to advantage.

In addition to proper joint preparation, the pressure welding machine must be operated with accuracy and consistency. The proper gas flows also must be maintained when making the weld. Paying attention to the above details will prevent the reported failure of "one joint out of every 50," as mentioned in the story on the pipe welding session at Detroit on page 86 of the September HPAC.—H. W. INSKEEP, development engineer, The Linde Air Products Co., New York City.

SHOULD LEAD TO WIDER STUDY OF CONDITIONING

It seems to us that the Safran Printing Co. approached the problem of air conditioning properly, in establishing its needs as described in the article, Printer Expects Big Savings Due to New Air Conditioning, which was published in the October issue.

We note with interest the estimated savings with the use of air conditioning equipment. Since this is a controversial item, it is rather difficult to make further comment. However, we are pleased that Mr. Safran is so satisfied with his system, which should encourage many others to investigate the matter and make further use of air conditioning equipment in their process areas.—B. OFFEN, B. Offen & Co., engineers-contractors, Chicago.

ARE WE BUILDING THE WRONG SCHOOLS?

While not a contribution to these pages, the following paragraphs certainly bring up a question that should be thrown "open for discussion." Readers are particularly urged to contribute their views for future publication, with particular attention to ventilation, heating, and air conditioning of schools.

"The taxpayer's dollar in most big U. S. cities is being spent for the wrong kind of schools; for buildings obsolete before construction even begins," and the average school today is "an extravagant monument to a dead past," according to the October issue of Architectural Forum, a special number on schools.

Foreseeing an expanded enrollment in U. S. public schools which will lift today's total of 21,000,000 pupils to an estimated 28,000,000 by 1956, the magazine reports that 400,000 classrooms must be added to the million now in use, at a cost to the taxpayer of $10 billion during the next decade. "With an education dollar that buys less and less," it says that public officials are faced with the problem of providing "more expensive equipment than ever before" to meet current educational standards of more space for each pupil, plus such extra facilities as work tables, individual libraries, medical and dental clinics, lunchrooms, etc.

To show how this problem can be solved, the magazine analyzes in detail 10 "of the best new schools to be found in the U. S.," pointing up the new departures which offer greater efficiency, frequently at lower cost. These schools offer such advantages as ample and controlled daylight for the child next to the window and the child farthest away; spacious classrooms which can be rearranged with the help of storage furniture; outdoor classroom areas; green or coral glass panels to replace the traditional blackboard and cut down eyestrain; Army-tested devices for audio-visual education.

Going beyond the best schools of 1949, the magazine also offers two hypothetical versions of the "1950 school." One is round, the other rectangular, but both these "schools of the future" offer the freedom and home-like atmosphere characteristic of the most successful contemporary schools combined for the first time with a radically new ultra-compact plan and with construction methods based on all the economies of factory construction.

A return to the one story elementary school building, even in the nation's largest cities, is one of the magazine's major suggestions for ensuring economy and efficiency in new schools. Citing the example of New York City, where new school buildings currently cost $18 per sq ft, the magazine maintains that this price could be cut 40 percent if New York were to take advantage of the economies of one story construction. Since these schools are extra-safe without costly fireproofing; since they eliminate the need for stairways, stair halls, duplicated toilets and extra exits; and since they can be built with light construction methods, the magazine

DO YOU HAVE something to say about any of the articles published this month? An additional comment, or perhaps a question on some point? We follow here the custom of engineering society meetings in allowing a period for discussion following presentation of a paper. You are invited to contribute your views. Address them to the Editor, Heating, Piping & Air Conditioning, 6 N. Michigan Ave., Chicago 2

"OPEN FOR DISCUSSION"

estimates that even on the most expensive land ever contemplated by the Board of Education in crowded Manhattan, a 10 room, one story school would save the city upward of $100,000 as compared with the same 10 rooms in a two story structure.

Highlighting a cycle which is becoming familiar all over the country today, the Forum points out that new residential communities and developments mushroom overnight, creating "a sudden overwhelming demand for grade schools and after that a somewhat lessened demand for high schools. And a few years after that, many of both kinds of schools stand empty."

"What is needed now," says the magazine, "is a realistic mental attitude which acknowledges that the school emergency is perpetual, and that there is no shame in so treating it. Instead of handling temporary schools as poor cousins, so badly built and maintained that the children rightfully hate to go to them, we should set out to produce temporary schools of top quality and performance."

Portable schools, which can be moved with population shifts, are another answer to this problem, suggested by a current experiment in Seattle.

Prefabricated schools, multiple use rooms, classrooms which can be expanded or made smaller with the help of movable partitions and re-evaluation of old standards for adequate lighting and ventilation are among the other possibilities detailed in the issue.

In many states, however, these improvements cannot be made, the magazine says, because of obsolete school codes. Some are "still based on ventilation theories proved erroneous in 1863," others are now "in direct conflict with what is known about the principles of providing good seeing conditions."

Converting school codes "from nuisances into useful instruments" is one of the five basic requirements predicated by the magazine for obtaining an up-to-date school. Others are: 1) fitting new construction into long-range plans; 2) reducing classroom types to a few high-standard basic patterns; 3) further developing prefabricated and mobile structures; and 4) taking advantage of important new trends in the overall school plan.

POLLUTED AIR WILL YIELD TO RESEARCH

Polluted air which assaults the city dweller's sinuses and eyes, dirties his clothes and curtains, and obscures his view of the sky is a costly nuisance that will yield only to scientific research.

Cleaning up the contaminated air of cities will require more than laws and good intentions. Atmospheric pollution is primarily a technical problem, and—until we gain enough knowledge about it through research—progress toward cleaner air will be slow.

We are on the threshold of important developments in the field because of recent research. Newly discovered facts about weather and the behavior of tiny particles suspended in air ,along with smoke abatement facilities and careful surveys of pollution in a number of cities, are aids to future control of atmospheric pollution.

There are three causes of contaminated air: natural sources, activities of individual citizens, and large scale industrial and commercial activities.

Volcanic dusts, salt nuclei from sea spray; combustion products of forest and grass fires, and sand or soil raised by storms are examples of natural contaminants.

It is likely that the presence of certain natural dusts and nuclei accounts for problems like Los Angeles' smog. What one city can tolerate in man-made pollution could be extremely annoying where bad weather and many natural nuclei are found in combination.

Information about wind, air turbulence, and the movement of atmospheric contaminants is important to city planners who can better choose plant sites and stack designs, while providing more open spaces and park areas.

Two ways to control pollution caused by individual citizens are: (1) good housekeeping and (2) foolproof equipment for such processes as fuel combustion.

Housekeeping involves simple precautions in such operations as handling and storing coal, building materials, and ashes and the burning of rubbish. Leaving rubbish or coal in the open, or carelessly unloading a truckload of sand are examples of poor housekeeping. Added together, thousands of such actions give the wind more pollution to blow about the city.

In windy cities, pollutions like soot and dust move from one part of the city to another many times.

As for equipment, present research is directed toward improved smokeless fuels and combustion equipment. A related field which has hardly been touched is development of smoke and cinder removal equipment for small combustion units.

Because industrial plants are an obvious source of pollution and accurate ways to measure their share have been lacking, they have often drawn more criticism than they deserved. Their pollution problem is probably the most amenable to eventual control.

Industries are eager to consider their neighbors as well as to avoid economic losses from the discharge of foreign material into the air. Through research, industry can learn about its pollution, integrate control measures with other technical requirements, and work out economically sound methods of control.

Constructors of industrial plants, for example, are now making careful studies of weather conditions at proposed locations before building.

Control of polluted air should come about through fundamental research and orderly technical development rather than through restrictive or punitive measures.—MORRIS A. FISHER, supervisor of fine particles research, Armour Research Foundation of Illinois Institute of Technology, Chicago (briefed from article in The Frontier, which is published by the Foundation).

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Vertical Blow Units Heat and Ventilate Paint Drying Oven

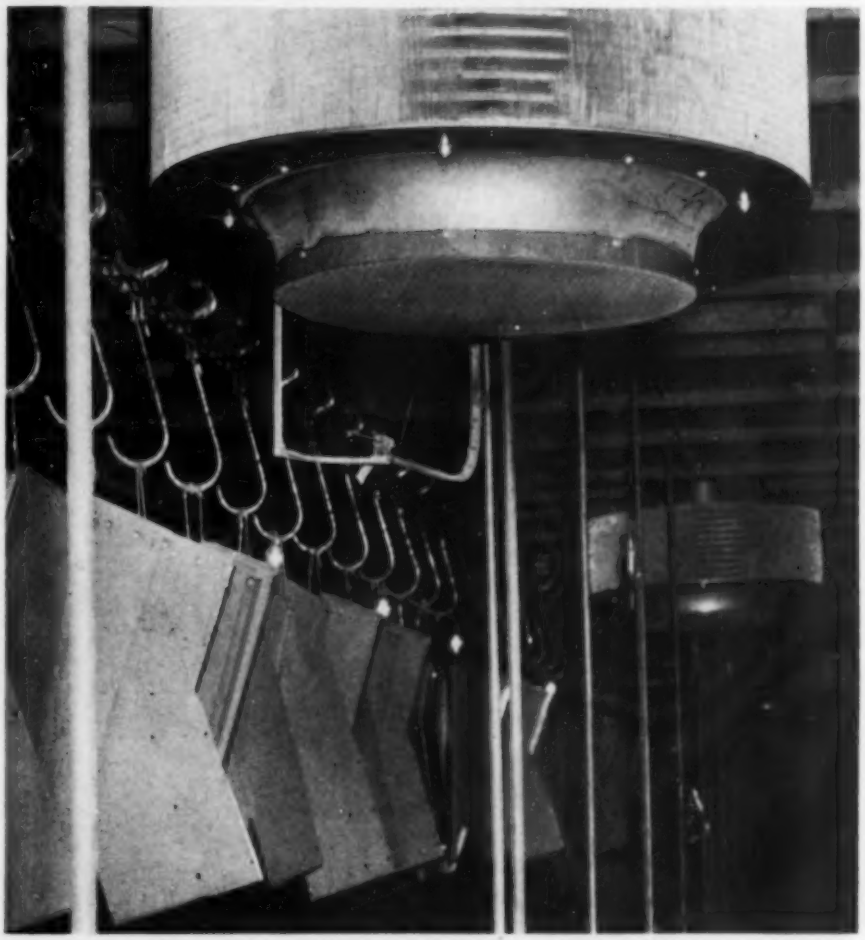
A WELL-KNOWN manufacturer of farm machinery, the J. I. Case Co., has recently completed and put into operation a unique installation of unit heaters in its main works, Racine, Wis. Five large vertical discharge propeller fan unit heaters are used to provide air circulation, ventilation air, and heat for a continuous, tunnel type, paint drying oven, completely eliminating all internal and external ductwork.

The oven is shown diagrammatically herewith. The main body is 108 ft long, 27 ft 6 in. wide, and 8 ft 9 in. high. It is mounted from columns 8 ft 9 in. above the normal floor level to conserve the space below for manufacturing and stock purposes. Walls, floor, and roof are constructed of No. 20 ga steel sheets screwed together on a welded channel-iron frame. The outside is covered with 3 in. of foam glass insulation on the floor and 2 in. on the walls and on the roof.

The oven ceiling construction embodies a number of loose panels which will blow outward in case of an explosion. The areas of these total 1740 sq ft, or about 60 percent of the entire oven ceiling. The oven is also "sprinkled" and inside heads are set up at 286 F. The heads immediately above the oven and under the building roof are set at 212 F.

The work is handled on a mechanized conveyor and comes from "Flo-Kote" painting equipment 75 ft away, entering the oven through a 20 ft long by 6 ft wide opening. Three passes are made before the work emerges from an exit opening of the same size. This requires a time cycle of about 35 min.

There are three unit heaters between the first and second passes and two heaters adjacent to the third pass. The paint is completely dry when reaching room temperature after leaving the oven, permitting decal nameplates to be applied immediately.



The oven capacity is 53,000 lb per hr of steel and/or iron farm machine parts. Forty gph of paint adheres to the average loading which forms the basis for evaporation requirements. Volatiles form the major part of the paint by composition and must be removed from the system for complete drying.

The Case laboratory specified a temperature of 180 F for proper drying of the paint types used. Because of explosion and fire hazard, ventilation as well as the application of heat is required.

Industrial insurance standards require that 10,000 cfm of standard air be exhausted per gallon of volatiles in ovens of this type. Approximately 4670 cfm of standard air at 70 F (or 5650 cfm at 180 F) is exhausted to meet this requirement.

The heat loss calculations for the oven and the heat required for drying of the paint were:

53,000 lb per hr of iron or steel parts to be dried	762,000
Heat losses through walls, floors, and ceilings	413,000
Heating make-up air for ventilation	675,000
Latent heat of evaporization for paint volatiles	40,600
Total heating load, Btu per hr.	1,890,600

HOWARD F. BRINEN, research engineer, Young Radiator Co., Racine, Wis., describes a unique installation of five vertical blow, propeller fan unit heaters which provide air circulation, ventilation air, and heat for a large, continuous, tunnel type, paint drying oven at the J. I. Case Co. main works.

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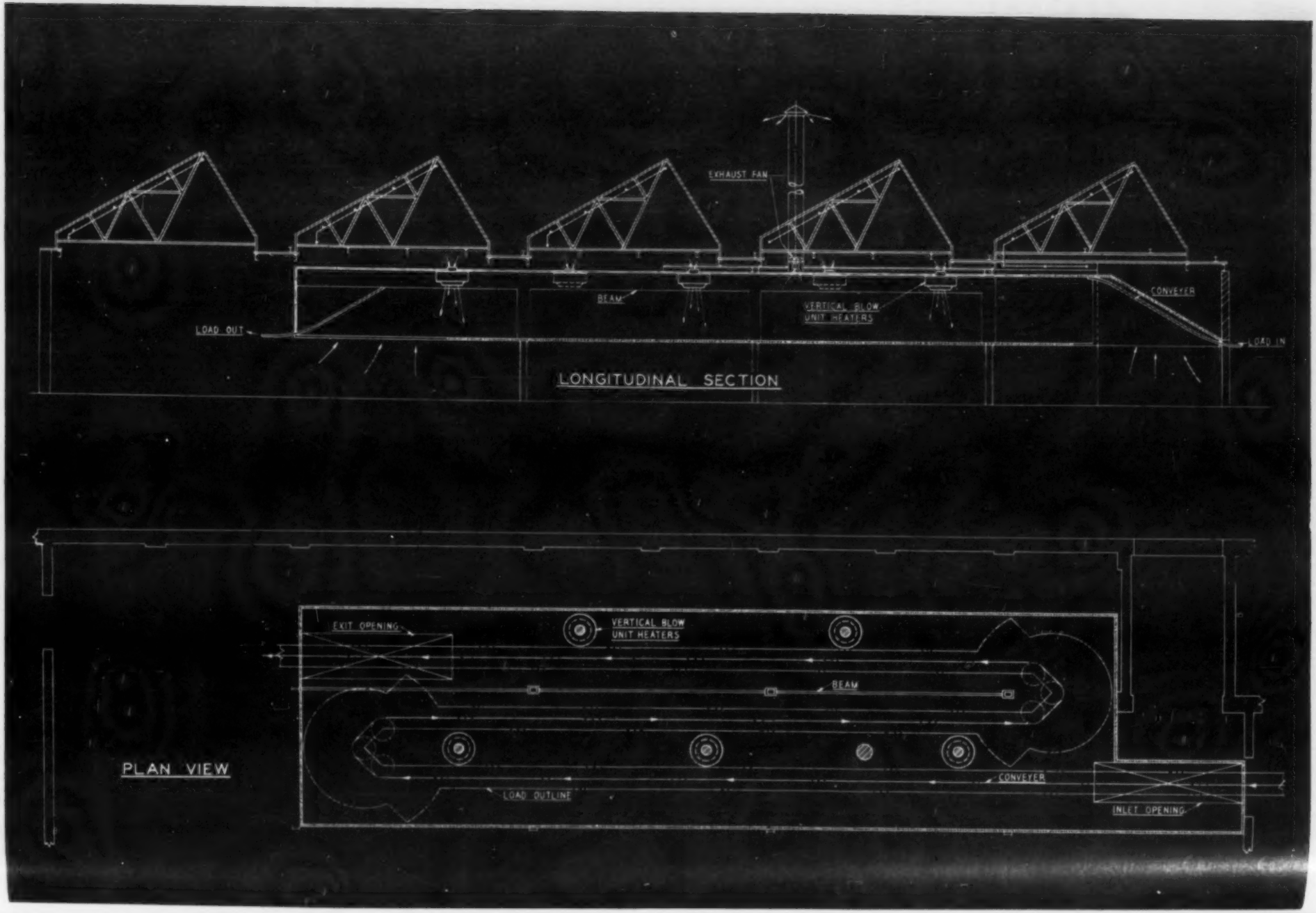

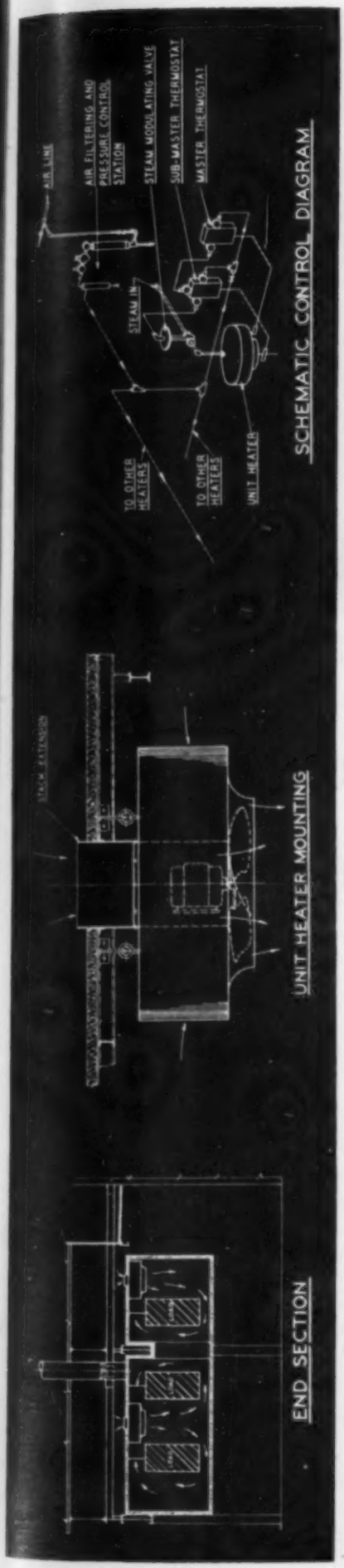

Each unit heater is rated at 550,000 Btu per hr at 2 psi steam pressure and 60 F entering air temperature. The oven can be operated with only three of the five units, and the total installed capacity of 2,750,000 Btu includes provision for future requirements. Since each heater delivers 11,380 cfm of standard air, the total air delivery of 56,900 cfm provides air circulation within the oven at the rate of twice the oven volume per minute, resulting in rapid and efficient paint drying. Of the 11,380 cfm per heater specified, 615 cfm is room air introduced through the stack-like motor mounting for normal motor cooling. A short duct section is fastened to each stack intake and is mounted as shown in an accompanying sketch to provide a portion of the oven ventilation requirements. The unit heaters are hung in the oven space on hooks, and an entire unit therefore can be removed in a few minutes.

Exhaust by Tubeaxial Fan

Exhaust ventilation is provided by a 24 in. diameter tubeaxial fan which exhausts through a 40 ft high stack in order to clear an adjacent four story structure. The capacity is 6000 cfm at 180 F, giving an air change about every 4 min in the oven. The unit heaters furnish a total of 3075 cfm of room air (3720 cfm at 180 F). This leaves 2280 cfm at 180 F required to balance the exhaust fan, and this is taken in through the entering and exit openings. This arrangement results in the oven being kept under a slight negative pressure and consequently fumes and hot air are prevented from getting into the working area below.

Insulation of Heater Motors

The unit heaters are powered by 1½ hp, 1100 rpm, constant speed motors. They are wound with Class II, glass type, electrical insulation because ordinary Class I synthetic type motor insulations can only be used on fan duty motors for ambient air temperatures which do not exceed 110 F-120 F. Factory room temperature air at substantially 70 F is drawn over the motors for motor ventilation during the normal operating cycle; however, shutoff of the oven brings the danger of high temperature and reverse flow through the stack, so special controls further protect the motors.

The normal blowdown capacities of the unit heaters range up to 50 ft. No diffusers are used in this application and the hot air streams blast directly against the oven floor, giving high velocities and unusual turbulence with resultant shorter drying periods.

Controlling the Unit Heaters

Controls are of the pneumatic type and each unit heater is individually and separately controlled. Starting and stopping is accomplished manually. The air temperature of the oven is controlled at 180 F by two thermostatic bulbs in the discharge air stream and at the side of each heater and which regulate a modulating steam supply valve on 125 psi line pressure. The oven comes up to operating temperature from a cold start in less than 5 min.

A second control system, electrically interlocked, provides high temperature protection for the unit heater motors along with other general safety features. When the steam supply valve for the heaters is opened, a pressure switch energizes the heater motor magnetic starters. When the same valve is closed, a thermostat keeps the heater motors running until the oven temperature has dropped to 110 F.

The exhaust fan is automatically started when the conveyor is started. It is interlocked with the unit heaters and operates until the heater motors shut off.

There are three dial thermometers of the remote bulb type mounted on the sides of the oven to indicate the oven space temperature.

The idea for this oven originated in the central plant engineering department of the Case company and design and construction details were worked out by the engineering staff of this department. Erection was by company crews. Cole H. Morrow is chief plant engineer of the company, and Max Wenscott is plant engineer of the main works where this unit heater installation is located.

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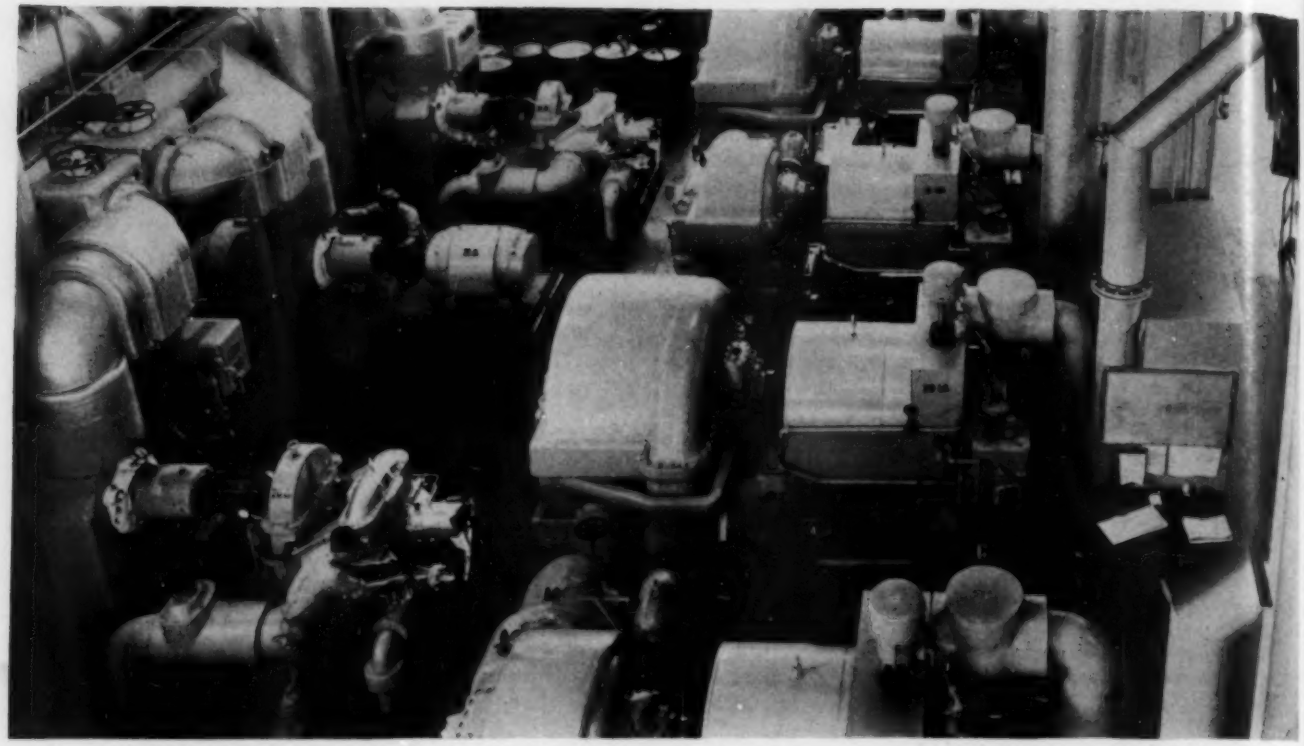

A general view of the operating floor, showing turbines, compressors, brine pumps, and piping

Minus 85 F Refrigeration for Film Processing at Kodak Park

A NEW refrigeration system with a capacity of 750 tons at —85 F is now in operation at the Kodak Park plant of the Eastman Kodak Co., Rochester, N. Y., and is in the process of being doubled. The system [we are advised] is one of the largest in existence at ultra-low brine temperatures.

The present refrigerating equipment consists of two 375 ton, "Freon 12," two stage units, each with two centrifugal compressors driven by steam turbines rated at 2325 hp per machine. Each machine occupies a building bay 22 ft by 36 ft with a basement 11 ft below the operating floor. Two additional duplicate refrigerating machines are in the process of erection with initial operation scheduled for 1950. With these four machines, a volume of 195,000 gal of methylene chloride brine will then circulate at 6900 gpm through nearly two miles of 4 in. to 18 in. nickel-steel piping.

Such equipment and the need for very low temperature refrigeration for the photographic industry was undreamed of 60 years ago, when the horsedrawn ice wagon was the standard. These progressive advances from 32 F down to —85 F represent research, experience, know-how, cooperation and initiative on the part of all who are in the midst of the refrigeration business—both manufacturers and users.

Back in the early 1880's, when the present Eastman Kodak Co. was struggling to make a start in the photographic field, ice was the best known and commonly used refrigerating medium. From 300 to 400 lb of ice was used daily in connection with the setting of photographic emulsions. As time went on, the refrigeration requirements grew, until today there is 28,400 hp installed for driving "Freon" and ammonia compressors at our Kodak Park plant. The drivers consist of up to 1500 hp synchronous 2300 volt motors and up to 1400 hp steam turbines. The actual summertime output of the plant at present is 750 tons at —85 F, 1200 tons at —36 F, 3725 tons at 9 F, 4400 tons at 35 F, and 1500 tons at 40 F. This total actual output of 11,575 tons is expected to increase to nearly 13,000 tons by the end of the current expansion program three years hence. Approximately 45,000 gpm of condensing water is circulated over 10 cooling towers.

Air Conditioning and Process Cooling

The 40 F, 35 F and 9 F systems are used for air conditioning and

special process cooling. The —36 F system using 1.29 specific gravity calcium chloride brine is used in two film production departments for the recovery by condensation of solvents given off in the film manufacturing process. The —85 F system, with methylene chloride as the brine, is rapidly replacing the 27 yr old —36 F system.

Film support, on which sensitized emulsions are coated, is manufactured in continuous operating machines. The film is wound into rolls 4 to 5 ft wide and 3000 to 5000 ft long. The raw material entering these machines is a very viscous liquid called "dope." The dope is a cellulose derivative dispersed in solvents. During the drying process in the machines, the dope gives off as a gas the solvents used in mixing the dope. These solvents are costly; therefore, extensive and complex equipment is installed to recover them for re-use. Part of this recovery process involves condensing the solvent gas in shell and tube condensers using —36 F calcium chloride or —85 F methylene chloride brine.

Study of Low Temperature Possibilities

About four years ago, an engineering investigation was begun to determine the possibilities of using some temperature lower than the old —36 F to condense solvents. This study involved brine temperatures ranging from —36 F to —100 F and various combinations of activated carbon adsorbers and scrubbers. The final report is very complex, involving 14 separate schemes and is outside the scope of this paper. However, some of the pertinent factors involved were as follows:

1) Five of the existing seven ammonia compressors on the —36 F system are steam engine driven, are 20 to 27 yr old, and nearing the end of their useful life. This fact was underscored when one of the compressors, then 23 yr old, wrecked itself in 1946 when the main shaft broke at one of the crank discs.

This machine and the adjacent unit both were replaced immediately by two "Freon 12" steam turbine driven, centrifugal machines complete with —36 F coolers and condensers.

2) A $25,000 order was on the books to repair cork covering and hangers on one of the two —36 F distribution mains. This order has been cancelled and only minor urgent work completed.

3) It would not be economical or practical to try to lower the operating temperature of the calcium chloride —36 F system as it would approach too closely the freezing point in the coolers and the viscosity at —36 F is near the limit for reasonable heat transfer and pumping cost.

4) A lower condensing temperature in the solvent recovery shells would cause more solvent to be condensed and lower the solvent concentration in the film machines. This would materially reduce the solvent gas loss out of the machines and improve the working conditions around the machines. The overall recovery of solvent would be much higher with —85 F than with —36 F brine.

5) Several of the old brine cooled solvent condensers would require replacement within the next five years due to age and inefficiency.

In brief, the recommended scheme showed the lowest initial cost, lowest operating cost, second lowest solvent loss, best ease of operation, lowest maintenance cost, reduced potential hazard in film machine areas, cleanest and constant humidity air to film machines, and least building space required for necessary equipment.

The present project, with four refrigerating machines and a third distribution piping system, is estimated today to total about $1,750,000. This figure represents the refrigeration plant and outside distribution lines only and does not include manufacturing department cost; it allows no credit for the use of the existing refrigeration plant building space and general plant power facilities utilized. Including one spare refrigeration machine, this four-unit project will have an installed capacity of 1125 tons.

JAMES H. SHAW, refrigeration engineer, power dept., Eastman Kodak Co., Rochester, N. Y., gives some "hot" information on a very cold subject here. He describes one of the largest systems in existence using brine at ultra-low temperatures—a system that is in the process of being doubled in capacity. He explains why this low temperature was adopted, selection of a suitable brine, and the search for the proper piping material

Investigation of Suitable Brine

Due to ultimate possibilities of operation, the piping system was designed for —100 F service. The refrigeration machines also are capable of—100 F operation at slight loss of capacities.

Because of the magnitude of the project requiring large quantities of piping, insulation, brine, etc., considerable investigation and research was justified. The first item to be settled was what should be used for the brine.

Due to our own peculiarities of requirements for a plant manufacturing film, safety and noncontamination of product were the foremost requisites of a satisfactory brine. Cost, of course, had to be a close second. The brine must of necessity have a very low freezing point and preferably a low viscosity and high specific heat.

Methylene chloride appeared to have outstanding favorable characteristics and few adverse properties and this material was selected as the brine. Methylene chloride is a colorless volatile liquid, is nonflammable under ordinary conditions, and is one of the least toxic of the chlorinated hydrocarbons. It is noncorrosive with metals and an excellent solvent for rubber, oils, paints, etc. Under certain conditions, if moisture is present, a weak hydrochloric acid will be created which is corrosive. At —80 F, the viscosity is 1.25 centipoises (water = 1.79 at 32 F) and the specific heat is 0.2705. The low boiling point of 103.6 F necessitated a closed system with surge or expan-

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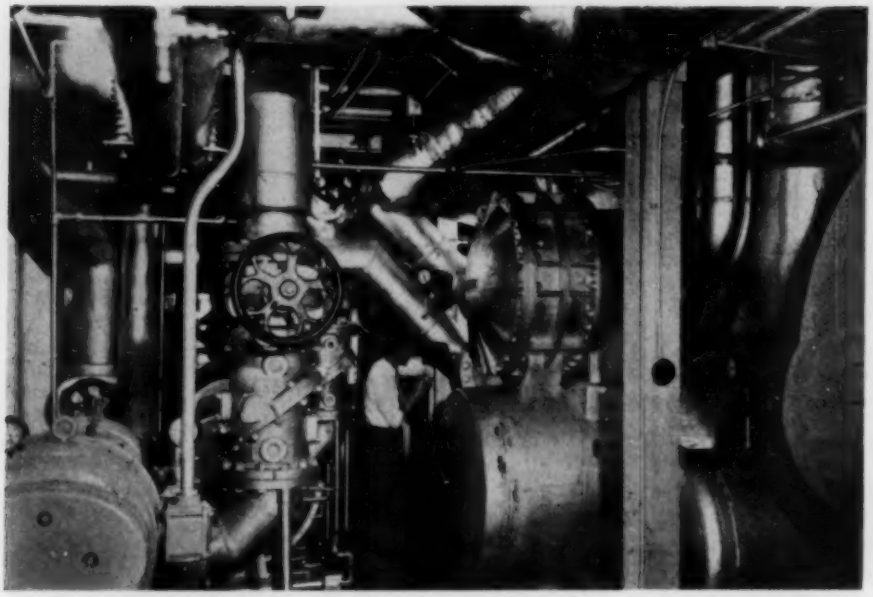


The author checks operation of some of the equipment in the big low temperature refrigeration system

sion tank. Two existing refrigeration installations outside Kodak Park have been using methylene chloride as a brine at —94 F successfully for about 10 yr. These two plants were visited and valuable information was obtained during the period of investigation.

Search for Proper Piping Material

Considering the large volume (approximately 195,000 gal) of the expensive (about $1.35 per gal) methylene chloride in the pipe lines and the low temperature involved, it was apparent that as nearly a leakproof system as economically possible was required. The search for the proper piping material consumed considerable time, with many inquiries for recommendations from piping manufacturers and fabricators and thorough testing by our own metallurgical laboratory. Our problem of selecting piping and fitting materials for —100 F operation required a critical examination of the ability of metals, whose room temperature characteristics were satisfactory, to resist the tendency to become brittle when used at —100 F.

Down to —100 F, the strength, endurance limit, and elastic properties increase with little change in ductility but with an accompanying drop in the notched bar impact values. In some cases the impact strength may drop off as much as 90 percent of the value at room temperature as follows:

SAE 1035 steel

Carbon, percent	0.36
Tensile strength, psi	79,000
Charpy impact values, ft-lb	
70 F	55
32 F	44
0 F	25
—25 F	15
—50 F	10
—75 F	8
—100 F	5

The standard notched bar impact (Charpy) test consists of a test specimen 10 mm (0.394 in.) square with 8 mm under the notch. The use of this test for acceptability of material does not necessarily insure that such material is acceptable for the service intended; that is, the notched bar impact test does not simulate service conditions of the equipment. If anything, it would appear that the notched bar impact test is more exacting, as the item usually is not so severely notched in service as the test bar.

Piping and fittings are usually subjected to repeated stresses in actual service, and the single blow of the test gives no indication of the notch sensitivity or lack of it. Passing or failing to pass an arbitrary foot-pound specification by a particular lot of steel carries no certainty of survival or failure of the piece in actual service. However, a uniform test, even though it be an artificial one, is necessary to give an indication of the character of the steel and is a criterion of the ability of a steel to withstand deformation not only in shock but in the static two- or three-dimensional stresses as well. Thus, the tensile strength, yield point, elongation and reduction of area, commonly measured at room temperatures, are of secondary importance to the impact test at the design low temperature.

For operating temperatures down to —75 F, special aluminum killed carbon steel with suitable heat treatment has been used successfully for welding fittings, forged steel flanges, etc., with consistent patterns of Charpy impact specimens showing values in excess of 15 ft-lb as required by the ASME unfired pressure vessel code. At around —75 F, however, some trouble has been experienced in getting the required Charpy values on the weld metal. For this reason the use of this steel should be limited to about —50 F, depending upon the experience of the fabricator, it would seem.

We planned to fabricate our own piping and fittings by welding and carbon steels were considered as unsuitable because of their inherent low Charpy values at —100 F. With some alloy steel next in line for suitability, the nickel-carbon steels were investigated.

Because nickel-steel obtains its desirable properties for low temperatures from its nickel content, its strength is not altogether dependent upon its carbon content as are the carbon steels. Since impact values decrease rapidly at low temperatures with increase in carbon content of the steel, the high strength of nickel-steel is obtained without the accompanying high impact brittleness as illustrated in the following table:

Nickel-copper-steel

Carbon, percent	0.10
Tensile strength, psi	78,000
Charpy impact values, ft-lb	
70 F	130
32 F	130
0 F	128
—25 F	125
—50 F	115
—75 F	95
—100 F	70

Nickel toughens steels markedly at low temperatures and its favorable influence is exerted alike on cast and wrought steels of any carbon content and in any condition of treatment. The addition of nickel increases the impact value with nickel content up to about 3.5 percent. The nickel creates a better initial room temperature resistance

to impact and lowers the temperature at which the steel begins rapidly to lose its impact value. Nickel also reduces the rate of impact value loss through the region of rapid loss.

Our metallurgical laboratory reported that all steels for low temperature should be thoroughly deoxidized with an excess of aluminum to produce a desirable fine grain material of 7 to 8 grain size with not less than 2 lb of aluminum per ton. Low carbon content is preferable, normally 0.16 percent maximum. Steels for low temperature service should be heat treated to produce a homogeneous structure. The treatment generally consists of heating to approximately 1600 F, water quench and draw at 1200 F.

The following table is typical and shows how all of these factors improve impact strengths. The low temperature tests were conducted at the University of Rochester:

	Charpy impact, ft-lb		
Material	70 F	—36 F	—85 F
0.16% carbon steel (boiler or tank plate)	46	6	2
0.16% carbon steel (aluminum killed)	55	11	4
0.17% carbon 3.5% nickel steel	46	34	28
0.17% carbon 3.5% nickel steel (aluminum killed)	51	43	40

Specifications for Piping Material

The piping specifications as finally drawn up and submitted for bidding were as follows: 3.5 percent nickel low carbon steel, seamless, high notch-toughness with Charpy value of not less than 15 ft-lb at —100 F, fine grain with McQuaid-Ehn grain size of 5 to 7. Chemical analysis as follows: carbon, 0.08 to 0.12; manganese, 0.30 to 0.60; sulphur, 0.04 maximum; phosphorous, 0.04 maximum; silicon, 0.10 to 0.20; nickel, 3.25 to 3.75 percent.

The piping was ordered from two sources, the 10 in. to 18 in. from one company and the 1 in. to 8 in. from another.

The 1 in. to 8 in. pipe as received and accepted by us was open hearth 3.5 percent nickel steel, essentially SAE 2317, and tested by heats which gave the following average properties.

Carbon, percent	0.10 to 0.17
Manganese, percent	0.45 to 0.58
Sulphur, percent	0.022 to 0.038
Phosphorous, percent	0.012 to 0.024
Silicon, percent	0.21 to 0.24
Nickel, percent	3.45 to 3.60
Yield point, psi	54,900 to 73,200
Ultimate strength, psi	81,200 to 89,200
Elongation in 2 in., percent	30 to 49
Rockwell 'B' hardness	81 to 85
Charpy impact, ft-lb at —100 F	39.5 to 44

The 10 in. to 18 in. pipe as accepted was open hearth 3.5 percent nickel-steel (SAE 2317), normalized at 1500 F, drawn at 1200 F, grain size of 5 or finer, with Charpy tests made on each heat before rolling. Following are the range of properties on all heats:

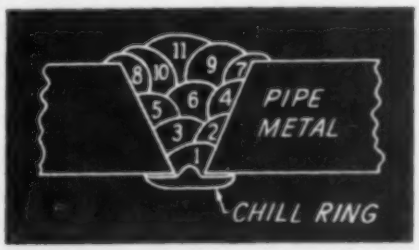
Carbon, percent	0.14 to 0.19
Manganese, percent	0.48 to 0.55
Sulphur, percent	0.021 to 0.035
Phosphorous, percent	0.011 to 0.027
Silicon, percent	0.18 to 0.30
Nickel, percent	3.15 to 3.47
Yield point, psi	56,600 to 62,250
Ultimate strength, psi	76,900 to 83,100
Elongation in 2 in., percent	34 to 40
Hydrostatic test, psi	700 to 800
Charpy impact, ft-lb at —100 F	37 to 46

Welding Fittings Made From Same Pipe

So that welding type fittings—such as elbows, reducers, etc.—would be assured of having the same chemical analysis as the pipe to which they were to be welded, the fittings were fabricated by an outside fabricator from nickel-steel pipe supplied by us from our large order. At the time of manufacture, this proved to be the most economical way of securing the steel for the fittings, since the amount of tubing required by the fabricator was comparatively small and this alloy steel was not in general production (which would make the cost of a small order of tubing excessive).

Both slip-on and welding neck flanges were forged from SAE 2317 nickel-steel billets. They were finished in the standard 150 lb ASA B-16e pattern. Test bars were cut from each billet, given approximately the same reduction in area as the finished flange received, were carried through the same heat treatment as the finished product, and then tested for Charpy impact at —100 F. These test bars gave Charpy impact values ranging from 37 to 56 ft-lb. One test bar was tested at —150 F and gave a value of 20 ft-lb, and one other bar was tested at —160 F and gave 37 ft-lb.

Welding diagram for nickel-steel pipe, using type 310 welding rod



All bolts and nuts used on the pipe line fittings are chrome-nickel alloy steel (18-8). These were not tested but were specified to conform to ASTM A-96-33 and to have a Charpy impact value of not less than 15 ft-lb at —100 F as determined according to ASTM E-23.

The chill rings to be used at all pipe welds were simply specified to be 3.5 percent nickel low carbon steel. This material (SAE 2317) was not tested as received.

Expansion joints are used at the suction and discharge openings of the four brine pumps. The joints have copper corrugations and nickel-steel flanges and trim and were designed for 100 psi working pressure at —100 F.

The gasket used at all flanged joints is an asbestos ring type flange gasket. According to the manufacturer, the material is suitable for methylene chloride, sulphur dioxide and other refrigerants. The gasket is made of asbestos fiber blended with superheat resisting compounds and bonded into sheet form under high pressure. After a year and a half of operation no evidence of leakage at any of the flanged joints is noticeable.

On the smaller piping (2 in. and under), wherever screwed joints occur, litharge and glycerine were used successfully with no joints found leaking to date.

Valves of 1½ in. size and smaller are solid bronze rising stem screwed type with asbestos rope packing. Gate valves 2 in. to 16 in. were specified and received as follows:

Nickel-steel, flanged, solid bronze wedge gate, nonrising stem of tobin bronze, 18-8 stainless steel seats, bronze stuffing box gland, tobin bronze stuffing box studs and nuts, 75 lb pattern with standard 150 lb

ASA B-16e drilling, nickel content to be 2.5 to 3 percent and carbon not over 0.25 percent.

The valves installed in the system include six 16 in. gates, eight 14 in. gates, eight 10 in. gates, four 10 in. checks, three 6 in. gates, 120 4 in. gates and 40 2 in. gates.

The valve body alloy as received is:

Carbon, percent	0.25 max.
Manganese, percent	0.75 max.
Phosphorous, percent	0.04 max.
Sulphur, percent	0.05 max.
Silicon, percent	0.20 to 0.40
Nickel, percent	2.25 to 3.0
Chromium, percent	0.70 max.

The tobin bronze stems have the following characteristics:

Copper, percent	60.25
Zinc, percent	38.91
Lead, percent	0.01
Iron, percent	0.08
Tin, percent	0.75
Tensile strength, psi	64,600
Yield point, psi	35,000
Elongation in 2 in., percent	41
Charpy impact, ft-lb	
100 F	16
0 F	18
—100 F	17
—200 F	16

The welding rod material and welding procedure used was determined by recommendations from outside fabricators and tests by our metallurgical laboratory. The rod used was 25-20 chrome-nickel, austenitic, mineral coated, type 310 in sizes 5/32, 1/8 and 3/32 in.

Welding Rod and Procedure

Some fabricators recommended this rod or a 3.5 percent nickel-steel rod, the pipe be preheated to 400 F and the weld stress relieved at 1100 F to 1200 F, holding for 1 hr, with either rod.

In our tests, various pipes were welded together with type 310 rod with no preheat and no annealing. Samples were cut from these joints and subjected to the standard face and root U bend tests. The bending tests indicated good quality of weld with no checks or other defects revealed. X-rays showed very sound welds free from all types of defect.

Due to thinness of specimens cut from welded pipes, only half size Charpy bars were cut. These were subjected to the impact test at low temperatures with the following results:

Sample	Location of notch	Temp. of test, F	Charpy ft-lb
B-1	Base metal	—90	28.5
B-2	Base metal	—87	28.6
W-1	Weld metal	—87	16.3
W-2	Weld metal	—89	16.0
H-1	Heat line	—80	34.4
H-2	Heat line	—89	35.4

As a result of the above tests, a welding procedure was adopted, using type 310, 25/20 chrome-nickel rod without preheat or annealing and making 11 passes or beads as shown in the accompanying sketch.

No. of passes	Rod size, in.	Welding machine setting Amp	Volts
3	1/8	90	22
8	5/32	125	24

On pipe 6 in. and smaller, fewer passes with 3/32 in. rod were used.

[In future articles, Mr. Shaw will describe piping construction, insulation, refrigeration equipment, control, brine pumps, auxiliary equipment, and preparation of the system for operation, and will present operating data.]

GIVES RESULTS ON ELECTRIC RADIANT JOB

In a recent summary of test results on three electric radiant heating installations, G. Gill Freyder reported results achieved in a third floor office. Electric radiant heating panels of conductive rubber are used on the third floor for supplementary heat, the lower two floors being heated by a steam boiler. Some heat from the lower floors of course escapes to the third floor of the building.

The building is three floors, brick walls, walls and ceiling fully insulated, and there are storm windows. The total floor area is 1600 sq ft. The accompanying tables give data on the ceiling panel layout and electrical consumption.

The heating in each room is individually controlled by a thermostat. During the night and on weekends, the thermostats are turned down. Sufficient heat filters in from the floors below to maintain temperatures well above freezing. The thermostats are then set for desired room temperatures when the offices are occupied.

It is difficult to compare the heating requirements of this office with other electric panel installations because this is primarily supplementary heating. The electric energy used from January to June last heating season indicates a seasonal use of 6600 kwhr for heating at a cost of $125.40. This estimate is based on an average heating season of 6300 degree-days and the customer's average cost of 1.9c per kwhr.

This application of electric resistance panel heating seems to be practical, as there was a considerable saving on installation cost. The operating cost, although higher than for a conventional heating plant using other fuels, does not seem excessive, and by evaluating the advantages of zone control, cleanliness, and comfort the additional cost can be justified, according to Mr. Freyder.

Ceiling panel layout

Lounge	108 sq ft	1,600 watts
Dining room	260 sq ft	2,750 watts
Washroom	14 sq ft	615 watts
Kitchen	98 sq ft	615 watts
Hall	127 sq ft	1,275 watts
General office	175 sq ft	2,225 watts
Private office	232 sq ft	2,720 watts
Private office	186 sq ft	1,915 watts
Drafting room	423 sq ft	4,960 watts
Total	1,623 sq ft	18,675 watts

Operating data

Month	Kwhr	Kw	Deg-days	Kwhr per deg-day
January	1,674	11.7	1,134	1.5
February	1,265	18.9	1,039	1.2
March	739	12.1	854	0.86
April	228	10.2	515	0.44
May	8	2.4	184	0.04

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Heating and Air Conditioning for a Modern Newspaper Plant

THE DALLAS Morning News building—opened last March and formally dedicated in May—is one of the most modern newspaper plants in the country.

The office section forms one part of the building and the plant or factory section the other. The division is not absolute but generally speaking is quite accurate; there are some offices in the factory section but no plant operations are carried on in the office section. The architectural and structural treatments of the two sections are different; the plant section is designed to take the greater weights of equipment and material and was not given the finer architectural treatment of the office section. This division of the building made it necessary that it be divided also for ventilating and air conditioning purposes.

Office and Factory Sections

The office building, excluding the basement, contains the executive offices, the reportorial, classified ad, circulation and editorial departments, the library and morgue, and the out-of-city receiving and sending departments. None of these departments presented problems that any other office building would not have. Two of the rooms presented the problem of shifting population load, which necessitated special treatment.

The plant or factory section of the building, separated from the office section by a corridor, is divided into closely related but wholly different departments. The paper storage space occupies a large share of the basement and extends over into the basement of the office building. The pressroom—largest room in the building—occupies about a third of the factory section of the building. The presses rest on the basement level or reelroom floor and extend well up into the first floor space. The first and second floor levels were omitted from this part of the building; in this area, the only break between the basement level and the third floor level is the press platform, or operating level, around the presses which is approximately halfway between the basement and first floor levels.

Most of the first floor area in the factory section is occupied by the mailing room. The second floor houses the complicated controls necessary for the operation of the

WHEN YOU'RE IN Dallas next January for the annual meeting of the ASHVE and the Southwest Air Conditioning Exposition, perhaps you'll keep in touch with world and national affairs by reading the Dallas Morning News. It is produced in one of the newest and most modern plants in the country. George Gregerson, chief mechanical engineer with George L. Dahl, architects and engineers, describes here the solutions to the heating, ventilating, and cooling problems of this building

presses, the dining rooms, the kitchen, the first aid room, and several offices. On the third floor is the composing room, the stereotype room, the job printing department, the photography department, and space for a future photo engraving department.

Convector Radiation and Unit Heaters

Heating of the building is handled by two low pressure steam boilers in the boiler room, which occupies a section of the office basement. The bulk of the heating is by means of convector radiation with unit heaters being used in storage spaces, machine rooms, and stair wells; ceiling mounted cast iron radiation in the workmen's toilets; and fan type convectors in the main lobby. Some of the spaces which have individual air handling units are partially heated by a split system with convectors handling the bulk of the heating and warm air the remainder. Because of lack of space for radiation of any kind, the mailing room is heated entirely with warm air.

The heating system is of the vacuum type, with the vacuum pump adjacent to the boilers. One part of the building is so far removed and at such a low elevation that it was found impractical to run the condensate back to the vacuum pump receiver. The condensate pump, vented into the vacuum system piping, is used to return the condensate from this location to the vacuum pump. Several parts of the building which lend themselves to zoning are controlled by means of thermostatically governed steam valves. All methods of steam distribution were considered but it was found that only an upfeed riser system would fit the building structure.

Most of the Air Is Humidified

Humidity control is very important in such a plant, and most of the air supplied to the various parts of the building is humidified. The air handling units throughout the building are provided with spray pumps discharging water through spray nozzles. These nozzles spray the water on to the coils and by so doing serve a dual purpose—they provide humidification as required in the winter, and a wet coil for summer use. For winter operation, the spray water is heated in an instantaneous steam converter. The air supplied to the basement paper storage area is humidified to 65 percent RH by means of a steam grid humidifier. Control is by a humidistat in the space to be humidified.

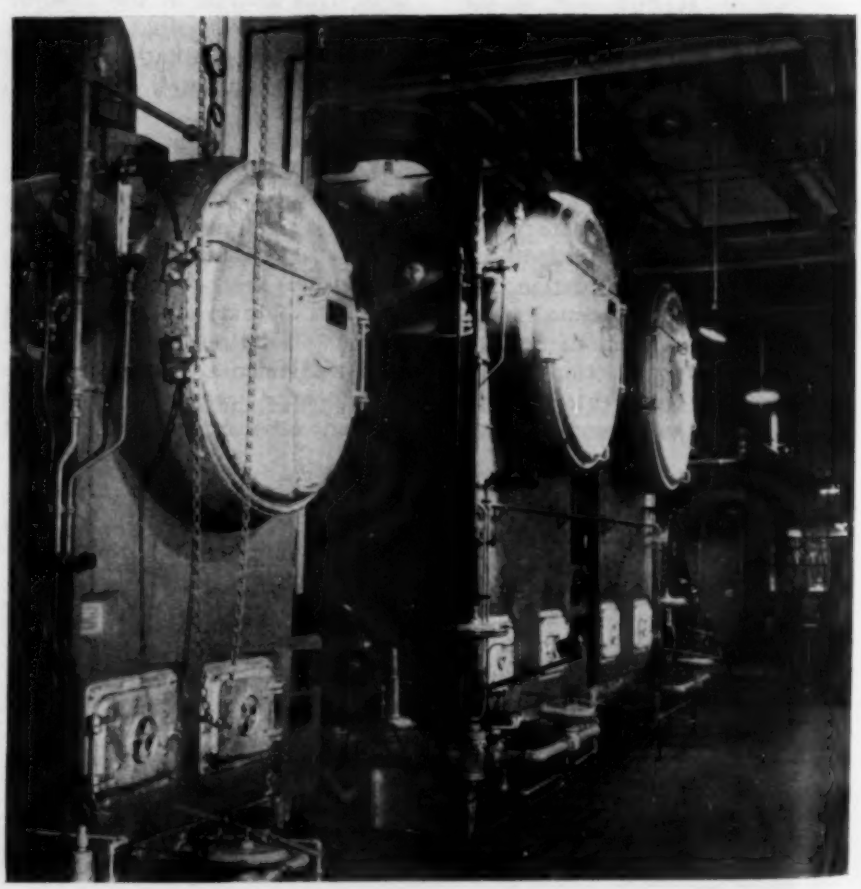
All parts of the building occupied by personnel are ventilated. Where feasible, the air is cooled. All office spaces, the composing room, dining room, first aid room, job printing department, wire photo, and photographic departments are air cooled. The storage spaces, machine rooms, boiler room, pressroom, press control room, kitchen, and stereotype room are ventilated as required.

Exhaust System for Intertype Machines

The gas consuming equipment in the composing room and stereotype department is provided with special exhaust systems. One fan handles the Intertype machines. Each of these machines has a flexible connection which discharges into a duct of larger diameter to eliminate flue gases from the burner and to pickup a small amount of room air. This pickup of room air serves two purposes. One is to absorb heat as it moves over and around the lead pot into the duct, thus preventing some of the heat from getting into the space. The other is to act as a diluting agent to reduce the temperature of the flue gases. The Intertype exhausts are connected into a common duct and discharged outdoors by a centrifugal fan.

The remelt pot, just off the composing room, presents two problems. The dross floating on the molten lead is burned off periodically. Before it is burned off, it is soaked with oil, and upon ignition, a dense smoke is generated which has a sickening odor. This necessitates that it be drawn off quite rapidly. In addition, the products of combustion from the remelt pot contain an adhesive substance which, upon cooling, settles out on the ductwork and on the fan. For this reason, a straight blade fan is used.

The boiler for heating domestic water (at left) and the two heating boilers. Bulk of the heating is by convector radiation, with unit heaters in storage spaces, machine rooms, and stair wells, ceiling mounted radiation in toilets, and fan type convectors in the main lobby



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The material making department, next to the remelt room, contains four machines which vary in design from the Intertypes used in the composing room to the extent that a fairly large hood is used for collecting the fumes and products of combustion. Because of the juxtaposition of the material making room and the remelt room, the two exhaust systems are combined into one fan.

The stereotype pot is heated by immersion heaters and is completely covered with a hood through which the lead and dross fumes and products of combustion are exhausted. This air, containing the products of combustion, is exhausted by a straight blade fan similar to that used for the remelt department. Here again, a substance forms which cakes on the inside of the duct and on the fan blades. A sufficient air quantity is used to maintain a ratio of about 12 to 1 between air exhausted from the room and the products of combustion.

Because of the generation of a small amount of undesirable fumes in the photographic department, a special fan is used to exhaust air from the developing rooms and the dark rooms. No special pains were taken to use noncorrosive materials because the fumes generated in the photographic department are not sufficiently corrosive to warrant noncorrosive ductwork and fan.

35 Fans Supply 211,805 Cfm

Other fans for supply and exhaust air in the building include: a boiler room supply fan used solely for the purpose of providing adequate combustion air for the boilers; an exhaust fan for the carburetor room to exhaust air in case a gas leak should occur (these carburetors mechanically mix air and gas to a combustible mixture, and pump the mixture to the stereotype pot and the flat casing pot); two exhaust systems for the toilet areas in the building; a special exhaust system for the carpenter shop for elimination of paint fumes; an exhaust fan for the transformer room in order to remove the heat generated by the transformers (these transformers increase the voltage from 208, the only one available, to 440, and this a-c current at 440 volts is rectified in ignitron tubes to direct current for driving the presses).

A total of 211,805 cfm is supplied to the building by means of 35 fans. Of this, 96,350 cfm is cooled by refrigeration and circulated through 11 fan systems. The remaining part of the air is ventilation air, which is supplied to various parts of the building and mechanically exhausted.

The refrigeration for air cooling is accomplished by means of two centrifugal compressors driven by 150 hp multispeed motors which are on the fifth floor in the office penthouse section. Two centrifugal pumps circulate the water chilled by the compressors through the various cooling coils throughout the building. Two similar centrifugal pumps circulate the condenser water through the condensers and to two cooling towers on the seventh and topmost floor. One of the prime reasons for locating the compressors and pumps on the fifth floor is that space in the basement is extremely valuable for other operations.

Zoning the Cooling System

The office section of the building is divided into three zones for air cooling purposes: east, central and north, and west. Because it was felt that the heat gain conditions on the north wall would be fairly constant, this area was included with the central zone. This raised the problem of how to obtain sufficient air changes in the central zone to prevent stagnation. Using a 20 F diffusion temperature difference meant that three to four air changes in the central zone would be sufficient to cool the spaces but not sufficient to provide adequate air movement. In order to increase the air movement in the central zone, all of the air supplied to the north or outer section is returned through the central zone. This increases the air movement sufficiently to prevent stagnation of the air.

To achieve flexibility in cooling, comparatively small air handling units are used for the various zones. Each floor has its own fan for the north and central zone. One unit handles the east zone for the first and second floors and a similar unit handles the west zone for the first and second floors. One unit handles the third floor east zone, with the unit so sized that it can handle the future fourth floor east zone. A similar unit is provided for the third floor west zone.

Four air handling units serve the factory section of the building. One, on the first floor, handles the mailing and associated rooms and doubles as a heating unit in winter. A second, on the third floor, handles the job printing department, future photo engraving department, and the AP wire photo department. A third, in the fourth floor machine room at the rear of the factory building, supplies cooled air to the dining rooms. All these air cooling units are of the package type, each "package" containing multiple centrifugal fans, cooling coil, heating coil, water sprays, and a filter box.

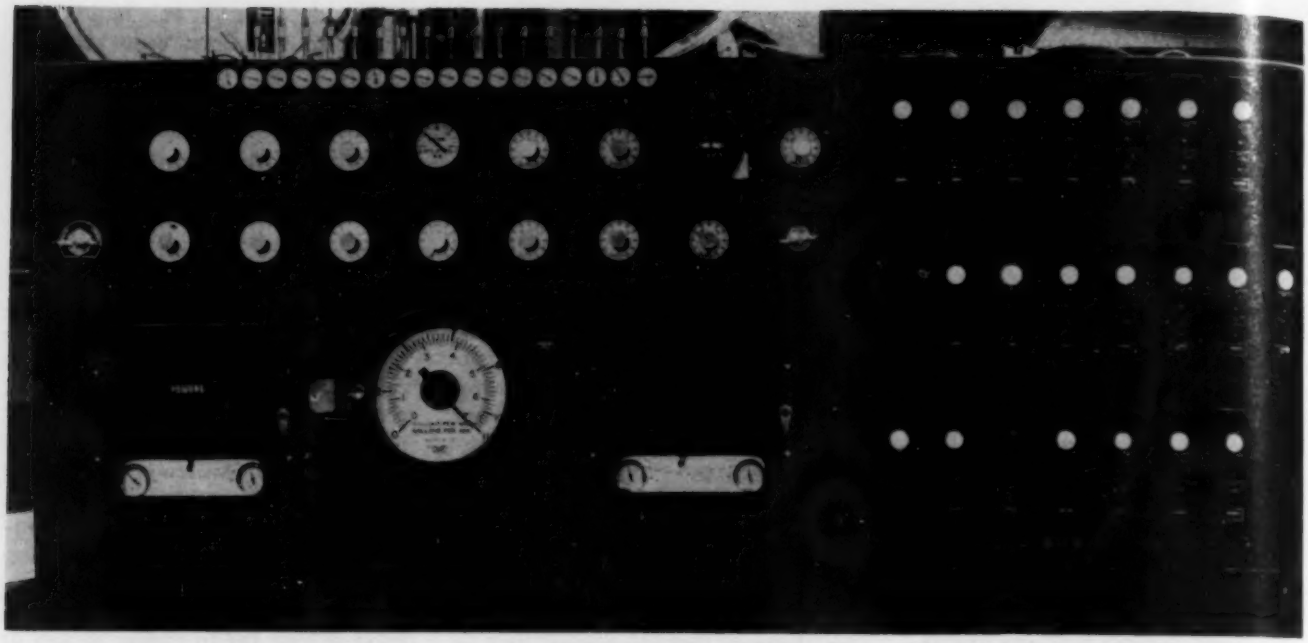
The composing room, having a cooling load far in excess of any of the other spaces, is provided with cooled air by means of a large double inlet, double width, centrifugal fan. A flooded coil section cools the air. A floor mounted, close coupled pump, just outside the coil section, circulates the water. Nonfreeze heating coils and an automatic air filter complete the equipment for this system.

All of the units handling cooled air are used during the winter to provide filtered and tempered air.

Because of the heat producing machines in the composing room—machines that produce obnoxious fumes—it was felt that even though they are vented, a large quantity of outdoor air should be continuously supplied. This air is not recirculated but is exhausted through the adjoining stereotype department, to provide—to a small degree—some benefits of air cooling. The cooling load for the stereotype department was found to be so high that it was not deemed feasible to air condition this space. Inasmuch as the air exhausted through the department is not sufficient to do a good job of ventilation, a separate fan in the fourth floor machine room supplies ventilation air.

Fan Supplies Outdoor Air to Units

Because the machine rooms for the zone units in the office building are so located that access to an outside wall for taking in outdoor air is difficult, a large fan on the fifth floor adjacent to the compressors supplies outdoor air to the



Central control panel for heating and air conditioning, during final phases of construction. (All photos by Dallas Morning News news staff)

air handling units in the office building. To enable the zone units to operate with 100 percent outdoor air, this fan is sized to handle 100 percent of the capacity of the zone units.

Inasmuch as the zone units normally use only 20 percent of the air supplied by this fan, the remainder goes to the basement ventilation fan. This basement ventilation fan serves as an exhaust fan for the office building as well as a supply fan for the basement. Thus, while the zone fans are using 20 percent of the air supplied by the outdoor air fan, the basement fan receives 80 percent of the capacity of the outdoor air fan.

Operation of Zone Fans

Outdoor air, recirculation air, and exhaust air dampers at the zone fans are so arranged that each zone fan individually may operate from a minimum of 20 percent outdoor air to a maximum of 100 percent. Thus, when the zone fans are using more outdoor air, the dampers permit less outdoor air to go to the basement fan and allow more exhaust air from the floors to go to the basement fan. It is therefore possible for the basement fan, if all zone fans are using 100 percent outdoor air, to get all its air from the office floors.

One purpose of the large air quantity used in the basement is to provide as much a feeling of coolness as possible for those working there. Inasmuch as this reason does not exist in the winter, provision is made in the control system so that during winter about 40 percent of the air supplied to the basement can be recirculated.

The basement fan provides ventilation for the entire basement. This air is exhausted through the press reel room and the pressroom by means of six vaneaxial fans in the upper west wall of the pressroom. Before the air is drawn into the fan, it travels through automatic oil type filters. These fans are, relatively speaking, quite noisy. However, they can be heard only when the presses are not running. Even then, there is sufficient attenuation in the room to keep the noise level below an undesirable level.

Few pressrooms are completely air conditioned. In most cases—as at the Dallas Morning News—all air supplied is exhausted outdoors. The fan discharge from the pressroom is so arranged that the air is traveling at high velocity to insure good aspiration and cut down the possibility of this air (although filtered) staining the outside of the building. The automatic filters are doing a fairly efficient job on this application but do not reach the efficiency in the removal of ink that they do on ordinary dust.

Schedule of summer temperature differentials

Indoor	Outdoor
72°	72°
73°	75°
74°	78°
75°	81°
76°	84°
77°	87°
78°	88°
79°	91°
80°	94°
80°	100°

Sliding Differential Between Indoors and Outdoors

A master submaster type of control is used for the regulation of summer temperatures in the building. By means of this, a sliding differential is obtained between the indoor air and outdoor air, as shown in the accompanying schedule. Temperature regulation at the units is obtained by face and bypass dampers. The thermostats in control of the dampers are reversible for summer and winter use. This changeover is made with a summer-winter switch. All temperature and humidity controls in the building are pneumatic.

The building was designed by the firm of George L. Dahl, architects and engineers. Technical data used in planning was obtained from Leven T. Deputy, chairman of the building committee for the Dallas Morning News, and William Ginsberg, consulting engineer, New York City. C. Wallace Plumbing Co. did all the mechanical work for the building. The Henry C. Beck Co. was the general contractor.

* * * * * *

Watch for the special section in the January issue giving complete information about the Southwest Air Conditioning Exposition to be held in Dallas January 23-27.

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Fig. 1—All parts of this river crossing header, except the 4 in. outlet at the bottom, were made of plate, shaped in presses and welded

Today's Pipe Welding Practices

F. C. FANTZ, vice president, Midwest Piping & Supply Co., Inc., St. Louis, has been intimately associated with the art of welding for some 40 years and was among the pioneers in its introduction in this country. His efforts have been an important factor in the growth of welding from its original status as a repair tool to its present vital position as a basic method of production and piping installation

PIPING SYSTEMS should not be thought of as composed solely of stock items of pipe, fittings and valves. Such "shelf goods" will always form a major portion of any line, if only for reasons of economy; but where operating requirements are beyond the adaptability of stock materials, the system should be designed to take advantage of methods to produce tailor-made sections of piping.

Welding is an easy, economical and effective method for the fabrication of special assemblies from formed and welded plate and their attachment in the line. It should be considered as a primary method of construction.

Advantages of Welded Structures

For example, Fig. 1 shows a design which has proved lower in initial and maintenance cost and greater in efficiency and safety than the awkward assembly it replaced. This structure is one of a number of river crossing headers for natural gas pipe lines. All parts of this assembly, with the exception of the 4 in. outlet at the bottom, are of plate, shaped in presses and welded. The tee section, formed to 26 in. OD, is very heavy material to compensate for the weaker section at the branch. The pad welded to the front of the tee is not a strength member but is applied to serve as a wearing plate to protect the tee from vibration against the concrete foundation. Elbow sections are reducing from 26 to 20 in. OD, thus producing a very efficient flow condition and reducing the stresses set up by changing of direction.

In contrast, note the assembly (Fig. 2) which the special design replaced. Seamless tubing was commonly used in all sections of this type of header, except for the caps and reinforcing saddles. It was necessary to apply the reinforcing saddles to compensate for structural weaknesses occasioned by insertion of branches. The flow restrictions and mechanical stresses are a great deal more severe than those in the improved design.

The assembly in Fig. 3 is a manifold for a steam bleeder piping installation. The tubular sections and the stock welding fittings comprising this assembly are of plate which was formed or rolled and welded. In this design, compactness and free flow conditions were major requirements. We, therefore, cut through welds and applied one section over another, so that in many cases welds intersect or overlap—a practice which has in the past been severely criticized by welding engineers. However, sound welding by competent welders under supervi-

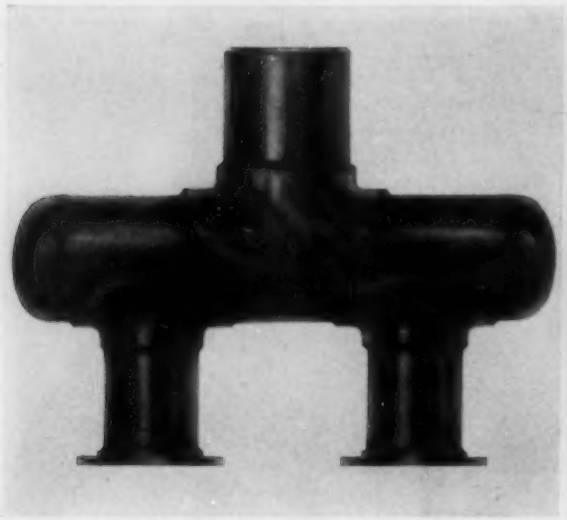

Fig. 2—This is the assembly that was replaced by the structure shown in Fig. 1. It was necessary to apply the reinforcing saddles to compensate for structural weaknesses caused by inserting branches

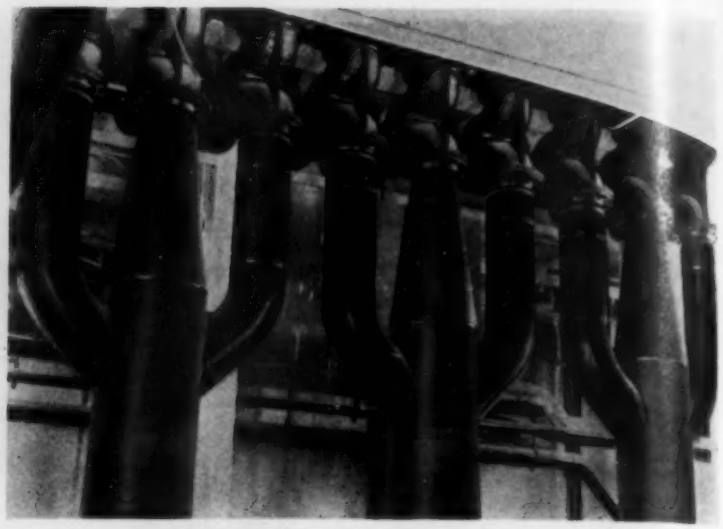

Fig. 4—Another example of what can be done to streamline and improve piping in congested sections. These assemblies are used to transport semi-solids, so good flow conditions are therefore most important

sion of an experienced organization cannot be questioned on the grounds of strength, in comparison with solid plate, seamless pipe or castings. If we believe in welding, then we should have no hesitancy in designing assemblies to meet the operating conditions encountered—and let the welds come where they may. The proof of this particular pudding is in the experience; this very rigid assembly—and many others similar to it—have been in operation for years without sign of weakness or failure.

Fig. 4 shows another example of what can be done to streamline and improve piping in a very congested section. This installation comprises sections of seamless pipe, stock welding fittings and special rolled and welded connections. The main section is 24 in. in diameter, with 14 in. branches reducing to 12 in. at the flanges. Since these assemblies are used to transport semi-solids, the advantages of superior flow conditions will at once be recognized. The valves in this installation are also of welded construction.

The compact and economical structure in Fig. 5 comprises a 20 in. welding elbow and two welded expansion joints for a catalytic cracking unit. Each expansion joint is composed of a double set of diaphragms welded together and capable of flexing in any direction. Bleeders or drains have been applied to the two sections that will be assembled in a vertical position.

Fig. 6 illustrates special fittings made in large quantities to meet two rigid requirements: first, the process demanded the best possible flow conditions, and second, the fittings had to be pressure-tight to a degree never before thought possible. All these fittings were fabricated from formed and welded plate and have met every requirement set up in the exacting specifications. Since great strength was not an important factor, no reinforcement was necessary; the fittings were made from the same thickness plate as the pipe to which they were welded.

One more photo (Fig. 7) illustrates the point. This mass of pip-

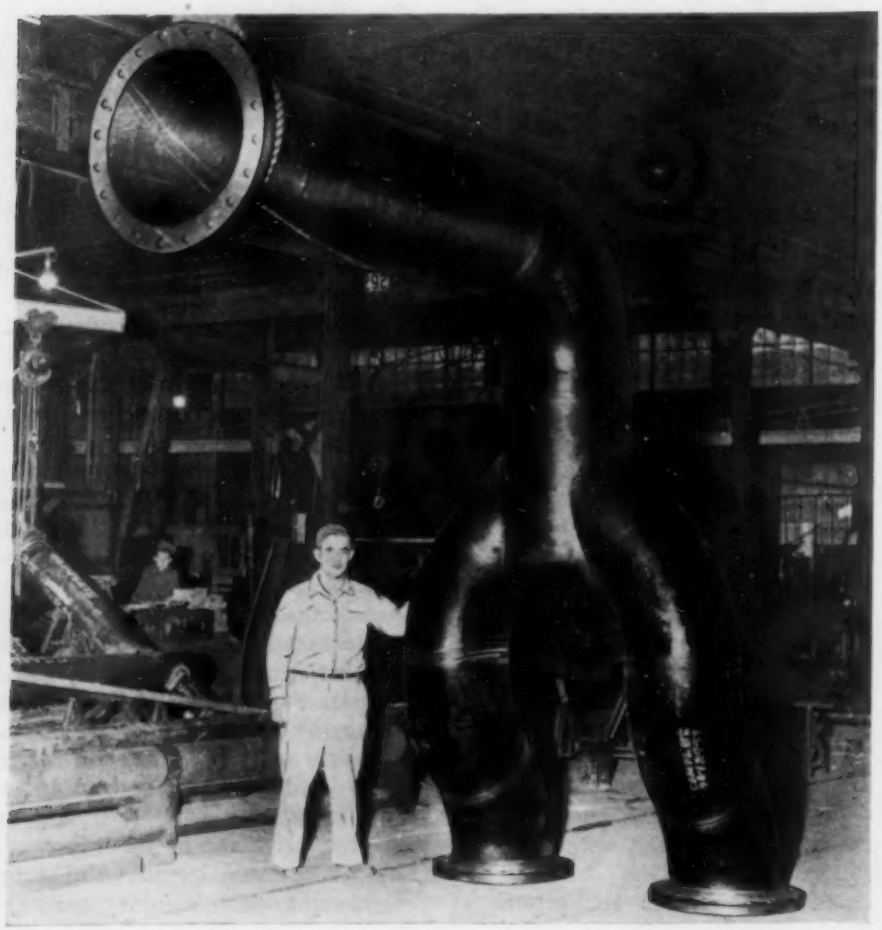

Fig. 3—In this manifold for a steam bleeder installation, compactness and free flow conditions were major requirements

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ing, fittings and connections is a perfect example of the space economies afforded by welded construction. The system incorporates almost all of the various types of piping connections in common use; butt welded and flanged joints; welding elbows, pipe bends and miter welded segmental ells; tees and welded nozzles; reducers and reducing elbows.

Types of Flanged Joints

The basic types of flanges commonly used in piping systems fall into three general groups:

1) Flanges welded to pipe, either by means of fillet welds or by means of butt welds;

2) Flanges applied to pipe laps, or lapped joints;

3) Flanges screwed to pipe.

The slip-on type of welded flange (A in Fig. 8) fits over the outside of the pipe and is double welded: a heavy fillet weld is applied between the hub of the flange and the outside of the pipe to act as a strength member, and a seal weld is laid between the end of the pipe and the face of the flange. This seal weld cannot be considered as contributing to the strength of the attachment, for the reason that failure must occur in the hub fillet weld before any stress can be exerted on the seal weld.

The various piping codes generally limit the slip-on flange to a 300 lb primary service pressure rating. However, there are many refinery installations in which this type of flange is used for all pressures and temperatures, and for pipe diameters up to at least 12 in. Such installations have been in service for 15 to 20 years without failure. This operating experience over a long period of time is admitted by code authorities to be above theory, and we can see no reason why this type of flange connection should not be more extensively used.

Fig. 5—Compact and economical structure comprising 20 in. welding elbow and two welded expansion joints

Slip-on flanges, particularly in 400 lb and heavier pressure ratings, should be refaced after welding to take care of distortion that is inevitably caused by welding stresses.

The most widely used type of flange is the high hub, butt welding, or welding neck (B). This has definite advantages over the slip-on type in that the distortion by welding is minimized and the design is superior in its resistance to stress. The butt welding flange, however, has the disadvantage that its inside diameter or bore must match that of the pipe to which it is attached. This can be a cause of delay, particularly where flanges for various pressures are used with pipe of several different wall thicknesses. The butt welding flange meets the requirements of all the existing codes for any pressure or temperature for which the flanges may be designed.

The third type, commonly known as a van stone flange (C), is used with a lapped pipe joint. The end of the pipe is lapped over to form a contact surface; the flange fits the back of the lap, and acts as a clamping device to pull together the abutting pipe faces.

The lapped joint has definite advantages: The loose flange can be revolved freely so that its bolt holes will line up with the facing flange on a valve or equipment. The pressure joint is made directly between the pipe ends, eliminating the chance of leakage between pipe and flange, and removing the latter from contact with the fluid in the line. (In many cases where corrosion alone is a problem and alloy piping is required, this permits the use of carbon steel van stone flanges with the corrosion resistant pipe.)

The screwed or threaded flange (D) has often been seal welded after installation. At best, this is only a makeshift, and should be used only for repair purposes. If the threaded flange is not sufficient to maintain a tight joint during operations, then welding can be employed to repair what is obviously a mistake in selection. The seal welded screwed flange, however, should never be selected as a means of producing an initial tight connection. The screwed flange is intended to withstand pressure and stress by reason of compression between the flange threads and the threaded section of the pipe. If we then apply a seal weld, we will certainly damage the original compression joint, and the weld immediately becomes not only a method of sealing but necessarily a source of strength. It should, therefore, be sufficiently heavy to withstand strains set up in the pipe line by temperature or pressure conditions, and the threaded connection, having been impaired, looses much of its effectiveness. From a design standpoint, it would have been wiser to select another type of flange, properly designed for the job.

Fig. 6—Special fittings made in quantity to meet best possible flow conditions and to be pressure-tight to a very high degree

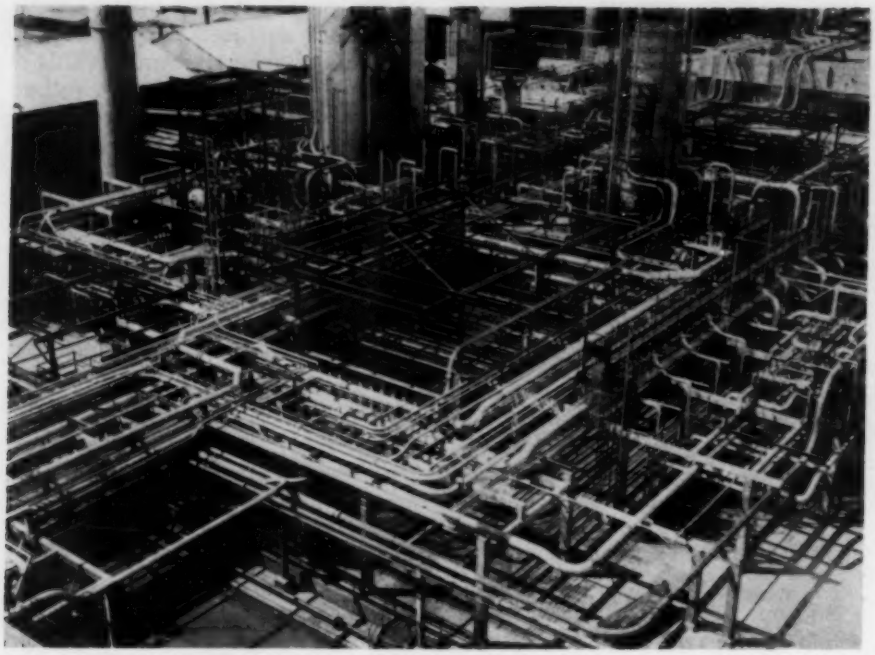

Fig. 7—This mass of piping is a good illustration of the space economies offered by welded construction

Fig. 9 shows the relative costs of the various types of flanged connections. These percentages are based on our experience and on current price relationships between stock fittings. Since costs and their methods of calculation differ widely, these figures should be considered only as a rough guide.

We believe it to be of special interest to note that costs of the welding neck flange and the slip-on flange are closely comparable. We have found that the higher cost of the butt welding flange is offset by the increased cost of installing the slip-on type.

Facings for Flanged Joints

Flange facings may be divided into two general classes, those which are applied directly to the flange regardless of whether the flange be attached by threaded or welded means and those which are applied to the lapped ends of pipe and used with standard van stone type flanges acting solely as clamping members. In either case the facings may be used in opposition; those applied to pipe laps are identical in dimensions with those machined directly on the flange or applied to valves and other equipment. Flange joints may therefore comprise an integral flange on a valve, and a lapped joint on the adjacent pipe; or a butt welding flange on one section of pipe may be set against a lapped joint on the continuing section of pipe.

Flange faces may be further divided in accordance with the means used to seal the joint. The most common method involves a gasket which is compressed between the two abutting faces. The primary purpose of such a gasket is to fill up the imperfections left on the joint faces in machining. Usually the gasket is of fairly soft material, such as asbestos or rubber. These have very little strength in themselves and flange faces are customarily machined with a low serration, in order to grip the gasket and prevent pressure in the pipe line from blowing out the joint. When the gasket is fabricated from a tougher material, such as iron or soft steel, the serrations will not be completely buried in the gasket, thus reducing the total gasket bearing surface and increasing the unit bearing pressure.

Types of flange facings are shown in Fig. 10, those utilizing flat gaskets being illustrated in details A, B and C. The raised face flange in A is the most commonly employed and is entirely serviceable for low and medium pressure work. This is the type of facing normally found on flanges drawn from warehouse stock.

Detail B shows a large male and female joint. The male section is identical with the ¼ in. raised face illustrated in detail A. The female section is provided with an annular ring for centering the flange and preventing gasket blowout.

In detail C is illustrated a joint with large tongue and groove facing. Here the gasket bearing-area is reduced materially to produce a greater pressure per square inch of surface. The gasket is confined in the groove to inhibit failure. This type of joint is an improvement over the old small tongue and groove type (not shown), which was considerably smaller in diameter and bearing surface. In the small joint, it was found that the male member under high bolting pressure would distort and mushroom, sometimes to the extent that the companion flanges had to be

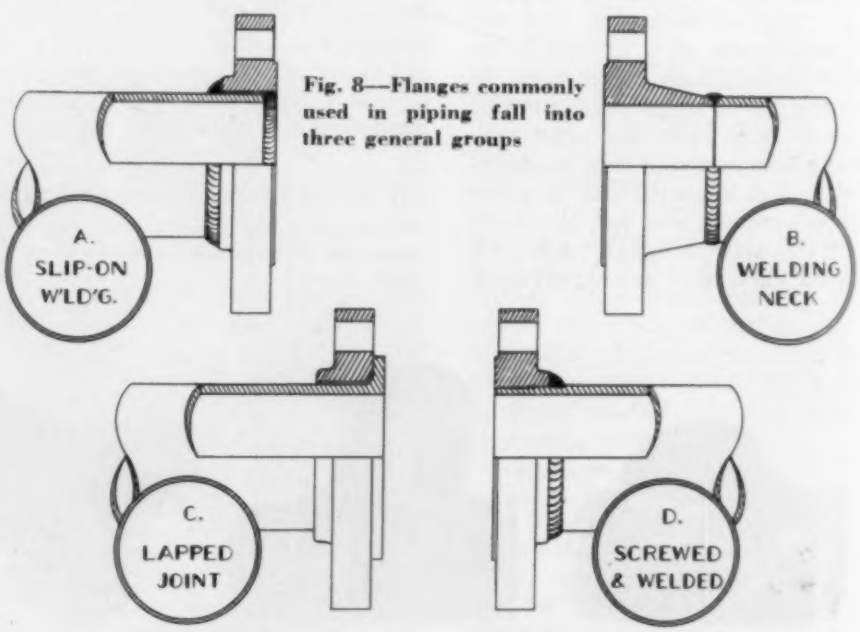

Fig. 8—Flanges commonly used in piping fall into three general groups

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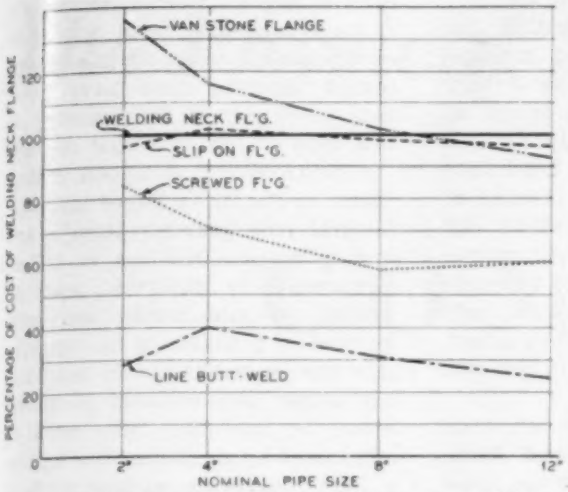


Fig. 9—Relative costs of flanged joints, the welding neck flanged joint being assumed as 100 percent. Costs include all necessary materials (flanges, bolts, gaskets, and welding rod), and labor (cutting, beveling or threading, lapping, welding or making-on, flange bolting)

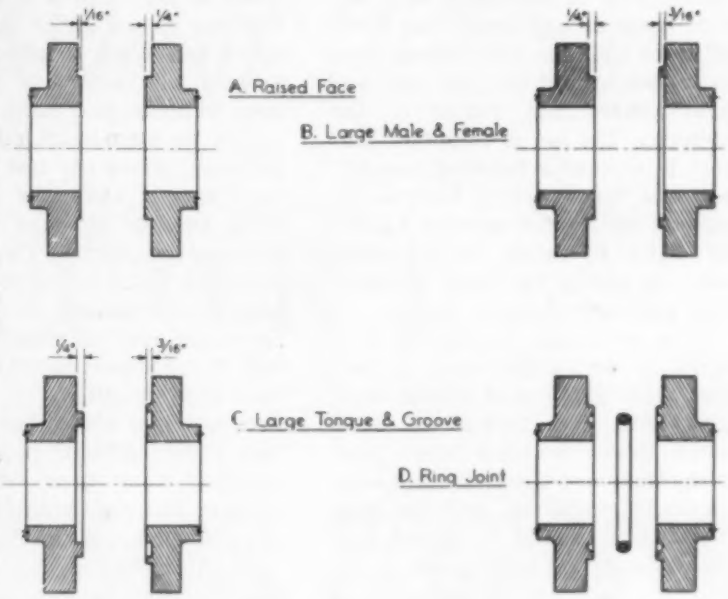


Fig. 10—Types of flange facings

forced apart to permit re-machining or dressing down of the tongue before the joint could be properly re-assembled.

In detail D is shown an entirely different type of flange facing known as the ring type joint. Here the gasket is in the form of a ring of oval cross section. This ring is of softer material than the flange, and of different shape from the tapering, flat bottomed groove in which it fits. A tight joint is achieved by applying sufficient pressure to deform the ring, so that it partially fits the taper of the groove into which it is forced. In recent years, this type of flange has gained in popularity, particularly for high temperatures and pressures.

If the tightness of a joint depends entirely upon compression between two flat members, any reduction in bolting pressure will have an equal effect on the fit of the gasket. In the case of a ring type joint, this is not true; only a portion of the reduction in bolting pressure is transmitted to the tapered ring gasket.

When a ring joint is used, it is customary to replace the gasket each time the joint is re-assembled.

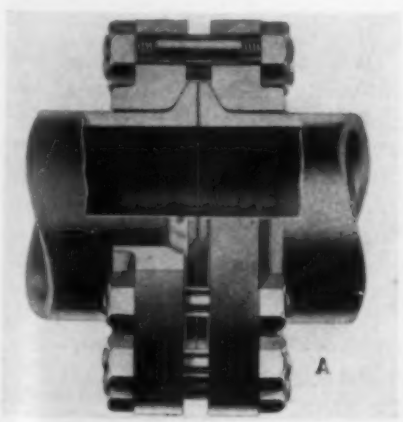
All of these facings can be applied either directly to the flange, or in the case of a lapped joint, to the pipe lap. In the latter instance, the flange itself will not change, since the gasket surface is machined in the lap rather than in the flange.

Detail A of Fig. 11 shows a "radial back" lap. As the name indicates, the back face of the lap is spherical. With the commonly used van stone lap, bolting pressures are limited by the thickness of the lap and the size of the fillet radius. Failures in the van stone lap, although rare, have always occurred through the radius toward the inside diameter of the pipe, as indicated in detail B of Fig. 11. To overcome this, the radial back lap was developed. At its outer edge, the thickness (after machining) is approximately two-thirds that of the pipe wall; at the base of the lap, this dimension is approximately one-and-one-half times the pipe wall thickness.

Special Lapped Joints

A comparison of details A and B shows the fundamental design difference between the radial back lap and the conventional van stone lapped joint. The improved metal

Fig. 11—The radial back lap type is shown at "A," while "B" illustrates conventional van stone lap. Failures (though rare) have always occurred through the radius toward the inside diameter of the pipe, as indicated in "B"



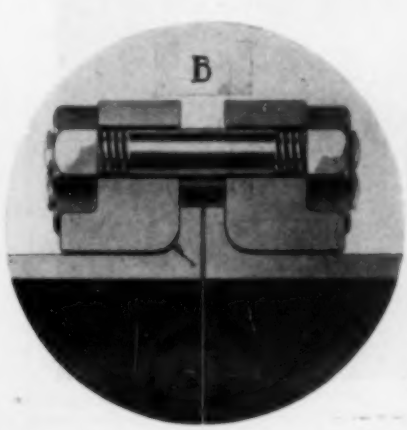
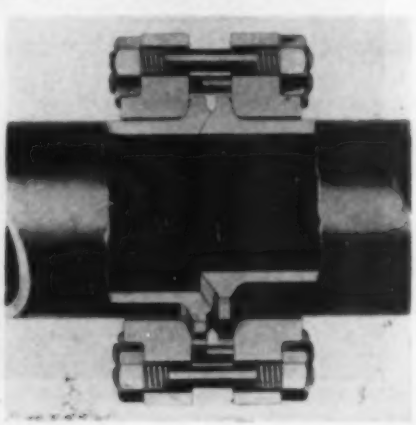


Fig. 12—A flared lap with integral ring gaskets, which has given satisfactory service in many high pressure, high temperature installations



distribution in the radial back design controls and limits the flexibility of the lap. The bolting force is shifted toward the pipe wall and acts toward the center of the sphere. The lap is therefore subject to a smaller bending moment, and the total parting force is reduced. Such joints provide a higher factor of safety, at no added cost, in piping for high temperature and high pressure service.

Some years ago, we carried out a series of destruction tests on regular van stone and radial back lap. Both were formed on 8 in. extra heavy seamless steel pipe; specimens were then cut from each, carefully machined, and the ends electrically welded to insure uniform tension on both pieces in the testing machine. Dimensions of flange bore, contact surface, etc., complied with the tolerances allowed in good piping practice.

When placed under tension, the specimens were strained in supposedly the same way as if they were in a complete joint, except, of course, for hoop tension due to fluid pressure. Since the test conditions were exactly alike, the absence of hoop tension did not affect the comparative results. The van stone specimen failed *in the lap* and separated completely at 56,970 psi; the radial back lap remained intact, and the specimen failed *in the pipe wall* at 58,480 psi.

It has been noted that van stone and other types of flanged joints removed from severe service show dishing and concentration of force along the edge of the flange or lap face. The construction of the radial back lap joint permits its use under similar working conditions without this type of distortion. In the last 20 years, literally thousands of these laps have been placed in service. In all cases, the laboratory results reported here have been fully borne out by the behavior of this type of lap in actual service.

A development of lapped piping shown in Fig. 12—a flared lap with integral ring gaskets—permits assembly without the use of either compression or distortion types of gasket. As said before, the only purpose of a gasket is to compensate for imperfections in machining, either unintentional or deliberate; in this joint, this need is eliminated by machining the laps to a highly accurate surface with a so-called "cold water finish."

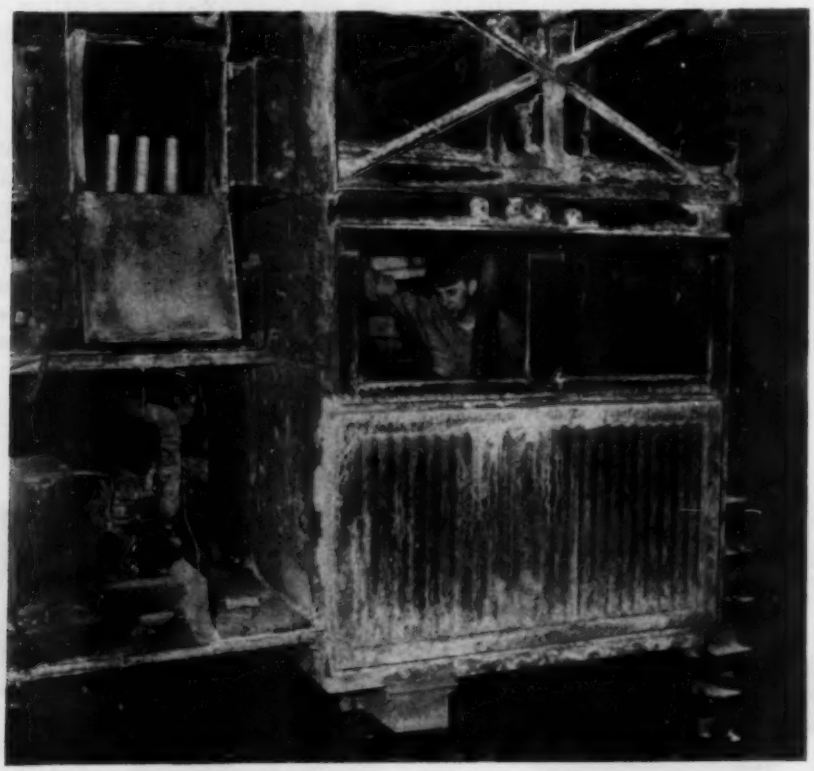
This type of joint utilizes male and female spherical surfaces on the lapped ends and a spherical contour comparable to the radial back lap on the back of the lap. Thus, it retains all the advantages of the radial back, and in addition, permits the union of two pipe ends when the centerlines of the adjacent sections vary in direction by as much as 3 deg. Discrepancies of this order can be handled without setting up stresses in either flanges or the pipe line. Another point in its favor lies in the fact that the assembly tightens as pressure and temperature rise.

In spite of its minor disadvantages which appear in the difficulty of assembling and disassembling this type of construction, and the care which must be used in handling the accurately machined faces, it has rendered satisfactory service in many high pressure, high temperature installations.

[A second article by Mr. Fantz will discuss directional changes in piping, reduction in line area, diversion or separation of flow, nozzle and weld reinforcement, welding procedures and welding processes.]

Good Air for Crane Operator

The air in this U. S. Metals Refining Co. smelter at Carteret, N. J., necessarily contains a certain volume of vapors and is usually at a fairly high temperature. An air conditioning unit maintains cool, even temperatures within the crane cab and an activated carbon air recovery installation bars the smelter fumes, thus giving the operator an almost ideal atmosphere in which to work. Access door has been opened and dust filter removed to show the carbon-filled canisters which soak up the gases. (Photo courtesy W. B. Connor Engineering Corp.)



Over 4000 current college and university research projects in engineering subjects, representing expenditures of over $35 million, are listed by title in the 1949 Review of Current Research, published by the Engineering College Research Council of the American Society for Engineering Education.

Copies are available from F. M. Dawson, College of Engineering, State University of Iowa, Iowa City, for $1.75.

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The Physical Side of

COMFORT

A. C. MENKE, and C. L. RINGQUIST, air conditioning engineer and manager, respectively, air conditioning department, The Trane Co., La Crosse, Wis., present a simplified discussion of the various factors affecting physical comfort. This discourse was prepared to clarify and explain the background and, hence, the results given in comfort standards widely used in the engineering field

LITTLE IS known about the stimulation or the development of the thermoregulatory mechanisms of the body. The center for the control of reactions to hot and cold has been located by physiologists in the hypothalmus region of the brain. Nervous control of the many mechanisms that play a role in the complete scheme of temperature regulation can be ascribed to three primary devices: (1) Thermal sensations may stimulate local effects which result in direct responses of temperature control mechanisms; (2) psychic influences may cause reflex adjustments of heat control mechanisms; or (3) stimulation of sensory nerve endings of the skin may cause impulses to travel to the spinal cord and brain heat control centers. The brain area responds to thermal changes of the body because of the impulses received from the sensory receptors of the skin and by changes in the temperature of the blood passing through the brain.

The sensations of hot and cold are practically limited to the skin and the lining of the mouth and pharynx. While some investigators believe that no individual temperature receptor organs exist but merely indefinite areas of greater or lesser sensitivity, most physiologists agree that there are many nerve endings in the skin which are separate in their response to heat and cold. The receptors for cold are more numerous than those for heat and are more readily stimulated.

Such subsidiary mechanisms as the adrenal and thyroid glands, and the blood concentration in the various organs of the body, are also a part of the overall plan of temperature control.

Partial regulation of body temperature by controlling heat production is carried out by chemical (change of basal oxidative reactions) and physical (muscular tenseness and tremor) processes but by far the greatest control of body temperature is accomplished by regulating the heat flow to the body surface. Heat flows from the sources of energy transformations to the skin surfaces by conduction and convection.

Internal Heat Flow

Neglecting the circulating blood, the conductivity of the skin and the tissues of the body is very low. The first thermal regulatory response of men on exposure to a cold environment is the diversion of the blood stream away from the skin by constricting the peripheral blood vessels. This provides the body with a considerable layer of insulating matter and, in effect, as described by DuBois [The Mechanism of Heat Loss and Temperature Regulation, by Eugene F. DuBois; Lane Medical Lectures, Stanford University Press, Vol. III, No. 4.], "converts the integument into a suit of clothes." If the thickness of the skin is taken as 0.15 in., then the conductivity with blood vessels constricted is 0.307 Btu per hr per sq ft per F, which is approximately the same as the conductivity of cork. Burton [Regulation and Blood Flow in the Human Fingers, by A. C. Burton; American Journal of Physiology, Vol. 127, 1939, p. 437] gives data showing that the range of blood flow in the fingers is from 0.5 to 1.0 cc per min per 100 cc of tissue as a minimum up to 80 or 90 cc per min per 100 cc of tissue for full dilation while conductivity varies between 0.30 and 1.30 Btu.

The skin conductivity is also affected by fat deposits. The layer of insulating fat between the skin and muscles is usually thicker in women than in men. Women can also more effectively reduce peripheral circulation and thus have approximately 70 percent [Basal Metabolism and Heat Loss of Young Women at Temperatures from 22 C to 35 C, by J. D. Hardy, A. T. Milhorat and E. F. DuBois; Journal of Nutrition, Vol. 21, April 1941, p. 383] more insulation in cold environments than men. In the cold, the insulation results in the attainment of lower surface temperatures (about 2 F lower than men), thus conserving body heat [Temperature Regulation, by J. C. Scott and H. C. Bazett; Annual Review of Physiology, Vol. III, 1941, p. 107].

The human body thus adjusts itself to low environmental temperatures chiefly by constriction of the

blood vessels of the skin and to high atmospheric temperatures largely by dilation of the blood vessels of the skin (along with sweat production).

Blood supply to the skin can be abundant and its concentration in any locality is controlled by the central nervous system. There are, especially in the extremities, many capillaries of the blood-vascular system forming networks throughout the body. A rich nerve supply is provided so that the skin can be flooded rapidly with blood or be voided of blood in accordance with temperature regulatory demands. The blood, consisting largely of water, has a high heat capacity so that a small temperature difference suffices for the flow of a large amount of heat to the blood in the interior parts of the body and away from the blood at the surface of the body. The body surface temperature depends upon the amount of heat dissipated to the environment and the amount of heat flowing by conduction and convection from the body interior, which in turn is a function of the blood supply to the peripheral tissues.

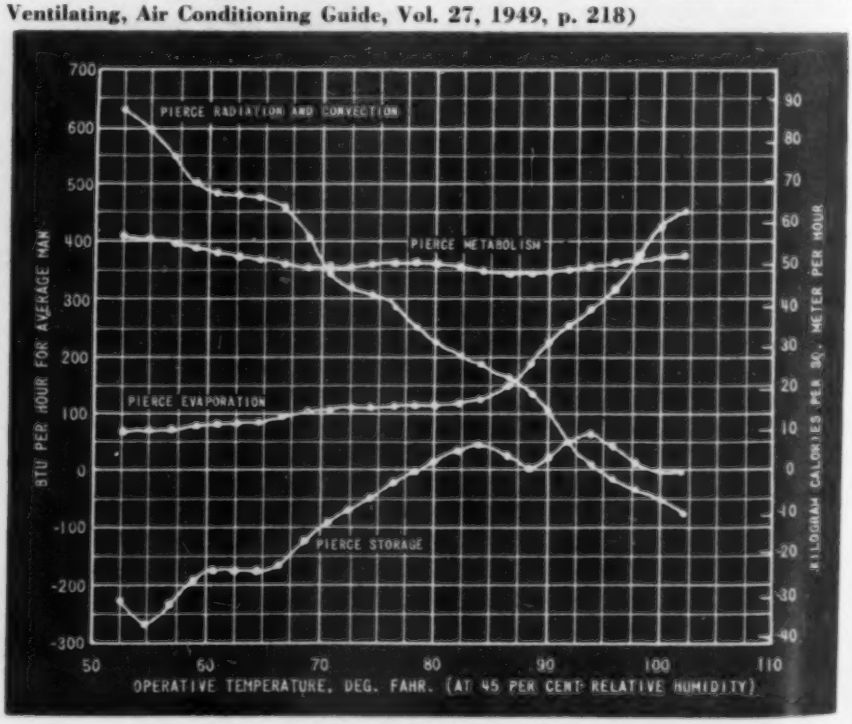
Keeton et al [The Influence of Physiological Research on Comfort Requirements, by Robert W. Keeton, Ford K. Hick, Nathaniel Glickman and M. M. Montgomery; ASHVE Transactions, Vol. 47, 1941, p. 159] reason that when the capillary bed in the skin dilates, the blood volume of the body evidently shifts from blood channels which serve as reservoirs to the vascular system of the integument; i.e., there is available a quantity of blood in quiescent storage depots which enters the circulation when the circulating volume must be expanded. According to this process, the total blood volume remains approximately constant but the circulating volume changes due to vasoconstriction and vasodilation. Winslow [Recent Advances in Physiological Knowledge and Their Bearing on Ventilation Practice; ASHVE Transactions, Vol. 45, 1939, p. 111] deduces that the body calls on its store of water in the body tissues to raise the total blood volume as more blood is required for circulation. Whatever the method of increasing the circulating volume, the observed rise can reach 8 to 12 percent in hot, humid conditions. This increase in the quantity of blood used to convect heat and, incidentally, to increase the conductance of the peripheral tissues, is the first adjustment to a hot condition. Cardiac output does not increase until there is a rise in temperature of the body mass or until the heat production of the body is affected. At this condition the peripheral vessels have reached a state of maximum dilation and further heat loss can be effected only by increasing the rate of blood flow.

Aside from temporary changes in circulating blood volume to adjust to immediate changes in environmental temperature, the control of the total human blood volume along with redistribution of blood between the interior and exterior portions of the body are important physiological adjustments associated with acclimatization to continued extremes of environmental temperature. Bazett, Sunderman, Doupe and Scott [Climatic Effects on the Volume and Composition of Blood in Man, by H. C. Bazett, F. W. Sunderman, J. Doupe and J. C. Scott; American Journal of Physiology, Vol. 129, 1940, p. 69] find a 15 to 40 percent increase in human blood volume in summer over winter. Large alterations in blood volume in response to climatic changes have also been reported by Taylor, Henschel and Keys [Cardiovascular Adjustments in Exposure to Dry Heat, by H. L. Taylor, A. F. Henschel and A. Keys; American Journal of Physiology, Vol. 139, 1943, p. 583]. The latter group conclude that the primary adjustment involved in acclimatization is an improvement in the cardiovascular efficiency.

For rapid adjustment to acute temperature changes rather than the slow modifications associated with acclimatization, the accurate regulation of the body temperature is accomplished largely by the peripheral vascular systems of the extremities. Benedict and Wardlow point out that 30 percent of the body heat dissipation occurs in the hands and lower legs which constitute only 12 percent of the total surface area. The circulation through an extremity can be, in high temperature environments, eight times that under comfortable conditions. In general the skin over the trunk, thighs and shoulders does not have the number of capillaries that respond to temperature regulation that are found in the skin of the hands and feet.

Summarizing, the flow of heat to the body surfaces takes place by the processes of conduction across the tissues and by convection as the blood flows from the hotter to cooler parts of the body. The physical regulation is due in the main to a change of conductivity over the

Relation between metabolism, storage, evaporation, radiation plus convection, and operative temperature for the clothed subject (from ASHVE's Heating, Ventilating, Air Conditioning Guide, Vol. 27, 1949, p. 218)



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gradient between environmental and internal body temperatures which is effected by control of the peripheral vascular system.

External Heat Flow

Heat production in the body and heat loss from the body must eventually balance if body temperature is to remain constant. Heat is lost from the body surface by radiation, convection, conduction and vaporization. Under ordinary circumstances a negligible amount of heat flows from the body by conduction and in calorimetric studies of humans, conduction is eliminated by using the least possible body support made of a material which is a poor conductor of heat. When no work is done on or by the body and conduction is neglected, the first law of thermodynamics applied to the body may be expressed by the following equation [Physiological Reactions and Sensations of Pleasantness Under Varying Atmospheric Conditions, by C.-E. A. Winslow, L. P. Herrington and A. P. Gagge; ASHVE Transactions, Vol. 44, 1938, p. 179]:

$$M = \pm S + E \pm R \pm C$$

where

M = rate of metabolism (heat produced in body)
S = rate of storage (change in intrinsic body heat)
E = rate of evaporative heat loss (always positive)
R = rate of radiative heat loss or gain; and
C = rate of convective heat loss or gain

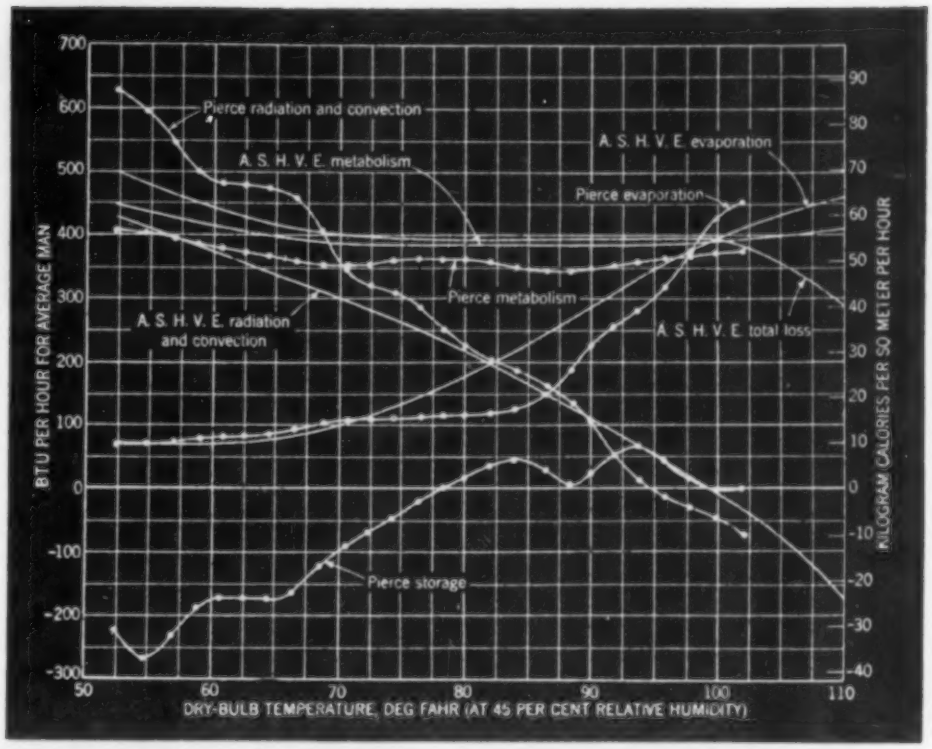
The important factors governing these heat exchanges are listed below.

I. Evaporation
  A. Environment
    1. Dry bulb temperature of air.
    2. Movement of air.
    3. Relative humidity of air.
  B. Body
    1. Moisture available for evaporation.
    2. Area of evaporating surface.
    3. Temperature of evaporating surface.

II. Radiation
  A. Environment
    1. Mean radiant temperature of surface.
    2. Emissivity of surface.
  B. Body
    1. Mean temperature of effective radiating area.
    2. Effective radiating area —shape factors.
    3. Emissivity of skin surface.

III. Convection
  A. Environment
    1. Dry bulb temperature of air.
    2. Movement of air.
  B. Body
    1. Temperature of effective convective area.
    2. Effective convective area.

IV. Storage
  A. Environment
    1. Total heat demand of the environment (as influenced by all the factors listed under I, II and III).
  B. Body
    1. Metabolic rate.
    2. Rate of external mechanical work performed.



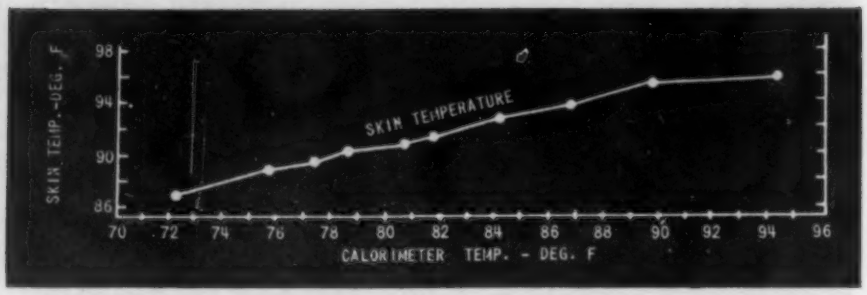
Thermal interchanges for clothed subjects at various dry bulb temperatures, comparing ASHVE and Pierce laboratory data (from ASHVE's Heating, Ventilating, Air Conditioning Guide, Vol. 26, 1948, p. 202)

Partitional Calorimetry

An analytical solution of the above equation is difficult because of the laborious task of establishing the values of the heat flow terms for the variable conditions to which the body is subjected. By making certain assumptions, however, Winslow, Herrington and Gagge, working at the Pierce laboratory, New Haven, Conn., have accomplished a solution.

To distinguish their method from the conventional form of calorimetry in which the measurement consists of a quantity of heat captured by the apparatus, they have employed the term *partitional calorimetry*. By this method the interchanges between the body and environment are determined as they occur instead of by summing results over a period of hours as in "heat trap" calorimeters. By partitional calorimetry, the results of heat transfer by the various physical avenues are determined by primary measurements associated with a particular avenue. Calorimetry of the ordinary type does not permit separate determination of the values of radiation and convection loss, whereas partitional calorimetry evaluates these effects independently. Furthermore, the process of adaptation to varying proportions of heat transfer by radiation and convection can be studied.

The general picture of the processes of thermal interchange as determined by the method of parti-



Importance of skin temperature in calorimetric studies (from ASHVE's Heating, Ventilating, Air Conditioning Guide, Vol. 27, 1949, p. 217)

tional calorimetry is shown in the curves on page 88. It will be noted that the temperature scale is in terms of the operative temperature index. This index is a generalized environmental temperature scale that combines in a single measurement any combination of mean radiant temperature, ambient air temperature, humidity and air movement. Its use as a generalized temperature scale gives experimenters in calorimetric studies a common reference measurement for diverse conditions. This factor is defined as "the equivalent environmental temperature with which a warm body (such as a human or animal subject) with surface temperature, T_s, exchanges heat at a standard cooling rate" [Standard Operative Temperature, A Generalized Temperature Scale, Applicable to Direct and Partitional Calorimetry, by A. P. Gagge; American Journal of Physiology, Vol. 131, Nov.-Jan. 1940-41, p. 93].

If the mean radiant temperature of the surroundings is equal to the air temperature, then the operative temperature is numerically equal to the dry bulb temperature of the air. This fact is utilized in the second accompanying figure, which compares the Pierce laboratory data on thermal interchanges for clothed subjects with ASHVE data.

Thermal Interchange Relations

The Pierce laboratory data are for semi-reclining subjects and the ASHVE data are for subjects seated at rest. All the information is for lightly clothed male subjects in still air (15 to 25 fpm) of 40 to 50 percent relative humidity. In the ASHVE studies, Houghten, Teague, Miller and Yant [Heat and Moisture Losses From the Human Body, by F. C. Houghten, W. W. Teague, W. E. Miller and W. P. Yant; ASHVE Transactions, Vol. 35, 1929, p. 245] calculated the total heat loss by measuring the heat produced and added to this the body storage, positive or negative, as indicated by changes in body temperature. They were able to measure the heat loss by evaporation and calculated that due to radiation and convection combined by taking the difference between the total heat loss and evaporative heat loss.

An accompanying curve illustrates the importance of skin temperature in calorimetric studies. From the nature of the factors involved, heat interchanges by radiation must be necessarily a function of the difference between skin temperature and the mean radiant temperature of the environment, and heat interchange by convection must also be a function of the difference between skin temperature and air temperature. Any deviations from a linear relation for the heat losses (or heat gains) by convection and radiation when plotted against operative temperature must be due to variations in skin temperature. Skin temperature at any point is the result of heat brought to it by convection and conduction and that lost by the various dissipative devices. Thus, the amount of heat to be lost from the body in order to maintain constant body temperature and the temperature and humidity of the environment are the chief factors which affect the surface temperature in various parts of the body. *Skin temperature control is the sensitive mechanism of the human thermal regulatory system which governs heat loss from the body.*

From the illustrations (Pierce data) showing the thermal interchanges between the *clothed* body and its environment, two critical points are observed. At 78 F (operative temperature) the heat produced by metabolism is balanced by the heat loss due to evaporation and radiation plus convection combined. At this temperature the body storage is zero. Beyond this point the body is warmed slightly until at 84 F (operative temperature) there is a definite inflection in the evaporation curve and the body begins to cool. The area between 78 and 84 F has been called the region of *vasomotor regulation* [The Influence of Clothing on the Physiological Reactions of the Human Body to Varying Environmental Temperatures, by A. P. Gagge, C.-E. A. Winslow and L. P. Herrington; American Journal of Physiology, Vol. 124, No. 1, October 1938, p. 30]. Below 84 F, the evaporative heat loss is essentially constant at a value associated with insensible perspiration. Above 84 F, the thermal balance of the body is maintained by increased evaporative loss as the heat loss by radiation and convection combined decreases until at 95 F the body actually gains heat by these avenues. The increased evaporation does not exactly compensate for the decreased radiation and convection, so that warming of the body tissues takes place.

The area above 84 F is called the *region of evaporative regulation*. Below 78 F, the evaporative loss is constant while the heat loss due to convection and radiation increases progressively as the environmental temperature decreases. Since this increased heat loss is not balanced by a change in metabolism or evaporation, there is a parallel cooling of the body tissues. The area below 78 F is called the *region of body cooling*.

[This is the second of a series of articles on this subject by Messrs. Menke and Ringquist. The first appeared in the October HPAC.]

MAKE IT EASIER

Maintenance and operation of mechanical equipment in plants and buildings are improved if concise instruction sheets or cards are mounted under glass in fan rooms, boiler rooms, and similar places. They should be brief for easy reading and to assure that they will be used.

Easy access to equipment through doors and openings of adequate size, and good lighting inside housings, also aid good operation and maintenance.

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Survey of Retailers Proves *Air Conditioning Essential*

THAT STORES and other commercial users in the 63rd and Halsted St. section of Chicago (one of the busiest and largest neighborhood shopping centers of the city) realized a dozen years ago the competitive advantages of air conditioning was brought out in the report of a 1937 survey by Knight C. Porter and William P. Rock published in the March 1948 HPAC.

The question is still an important one, and last May the author made a survey, comprising direct personal interviews with over 150 store proprietors and managers, in the same area. The questions asked were:

"Does air conditioning increase your summer business?"

"If so, why do you feel that way?"

Or in the case of a non-air conditioned store:

"Why haven't you air conditioned your store?"

"Do you think it hurts you competitively?"

A "Must" for Restaurants

In 1937 there were no air conditioned restaurants in the area surveyed. Today, there are four. (Grills and sandwich shops were surveyed, but only places with full-course menus were considered as restaurants.)

Why? Well, one proprietor laughed when he was asked. "To stay in business, of course!" was his reply. Another restaurant manager remarked that it was a "must" in the restaurant business. "The public demands it," he said. Another restaurant proprietor, when asked if air conditioning increased his summer business, replied, "It does, and I have the figures to prove it." This restaurant has been in the same location for over 15 years, and did not install air conditioning until 1939.

Not every retail store manager was as emphatic about the necessity of air conditioning as were the restaurant men. However, many stores have equally impressive records. In 1937, for instance, there were three air conditioned shoe stores out of 10 establishments. Today, in the same area, there are eight air conditioned stores out of 11 establishments.

Sells More Shoes

The manager of a large chain shoe store remarked that one of the important phases of air conditioning is keeping the salesmen comfortable as a direct benefit in the selling of more shoes. "Personnel can't do a good job of selling if they are uncomfortable and irritable," he said. The proprietor of a large medical shoe store merely said he installed air conditioning when he moved to this new building. He had been located in the area for many years but had never air conditioned. It is interesting to note that he wasn't certain of the competitive advantage of air conditioning, yet the drive to have a new location — with a new front and new fixtures —was all part of the desire to keep up with the other shoe stores on the block . . . and all the shoe stores on his block have air conditioning.

The most competitive line in this area is, I believe, women's apparel. These stores not only dominate the field in numbers but appear to have the largest and also the finest establishments.

The manager of one of the pioneers in air conditioning in this type of store replied to my question, "Why, mister, air conditioning *saves* our summer business. We would lose all the benefit of our advertising and whatever traffic we could attract into the store if we didn't have air conditioning."

However, he also made a distinction concerning various store sizes in relation to air conditioning. The larger the store the more important it is that it be air conditioned, he said. A smaller store, even women's ready-to-wear, possibly can get away without air conditioning. But a large store can't do without it. This was his analysis, and managers of other large stores made similar comments.

A manager of a large chain store was very worried because her air conditioning system was not on, and the service company couldn't promise her service for quite a while. This was during an early May heat wave and right before

IRWIN H. DIAMOND, of Air Conditioning Service, Chicago, has a two-way interest in store installations—the first being sales and service of air conditioning equipment, and the second due to a course at Northwestern University on retail store management which he completed earlier this year. For his term paper, he surveyed commercial users of air conditioning in a neighborhood shopping center which had been studied by others in 1937 and reported on in an article in our March 1938 issue. Some of the results of his informal survey are given here

Mother's Day. She painted a rather gloomy picture of her Mother's Day sales if the heat kept up.

The manager of another large chain store reported that customers refused to try dresses on during the summer months at one time when their air conditioning system broke down. "They just walked in and walked out," she said. She felt that people search out the store with air conditioning.

Two proprietors of small size, independent, women's apparel shops that were not air conditioned said that air conditioning was important, but they felt it cost too much. Both, however, said they have been giving future installations serious consideration. One felt he was losing out because so many of the other stores on the block were air conditioned.

No Millinery Users

This illustration of the importance of air conditioning in retail women's apparel shops is interesting when compared with the millinery shops in the neighborhood. Four shops were interviewed; none had air conditioning. All the shops had a nice traffic flow, and outside it was hot and humid. Yet, none of the shop managers interviewed listed air conditioning as an essential. It would be nice, they would like it, but certain characteristics of the business made them feel it wasn't required.

First of all, July and August are not in the millinery season—their season is over by June. The hot spell in early May this year was exceptional, but it didn't seem to be hindering the millinery business. Another factor is that none of these hat shops is very large.

Since 1937, there has been a tremendous business growth in this area. Specialty shops have sprung up and the chain stores have become more entrenched. An art shop that moved from one location to another considered air conditioning on a basis that has set a pattern of thinking in many jewelry and similar shops. That is, if the consumer would spend more time in the store, they would tend to increase impulse buying.

Comfort Aids Impulse Buying

Retail Management magazine (November 1948) reported the following on impulse buying:

"Merchandisers who have studied the subject from every angle maintain that customer comfort will contribute enormously toward increasing the number of unit sales. This is because the comfortable shopper is a leisurely shopper and leisurely shopping creates impulse sales. The importance of impulse buying is readily apparent since this has been found to account for 38 percent of all retail purchases."

A long established jewelry store reported that the customer will come into a store whether it is air conditioned or not because of the reliability of the store and reputation of the owner. This reasoning was by no means singular, for only four of the nine jewelry stores were air conditioned, and even the air conditioned stores did not go so far as to say that air conditioning increased their summer business. They all felt that it helped, but wasn't too important.

However, one large chain credit jeweler reported that when people come in to make payments they linger around a little watching the television sets, etc., because it is comfortable. Frequently, some other merchandise attracts them, and ultimately they purchase it. What the frequency of this type of sale is or the relative dollar value of the sale is unknown; however, occurred frequently enough for him to mention it, which in itself is indicative of its importance.

The proprietor of a small liquor store made an interesting comment. His store was not air conditioned; however, he did not have any competition for blocks around right "on the avenue." He wasn't particularly concerned about air conditioning, although he would like it. But when asked if another liquor store opened up across the street and had air conditioning, would he then install it, he replied, "Immediately."

There are not many grocery stores in the area, but the manager of a large "super-mart" that was planning to remodel within the month reported that air conditioning was included in the plans. He also said that customers often remarked about the shopping comfort at a nearby competitor's. This store is very large, accommodates many at a time, the large plate glass windows face the south, and it is a one story structure. All this contributes to the heat load of the store, and may have had a lot to do with deciding upon air conditioning.

Lease Factor Is Important

An important point in considering air conditioning is the lease factor. This problem has been more apparent since the war, and continues to be a bone of contention with many retailers. One confectionery store owner expressed concern over his rental arrangements by saying, "I would buy air conditioning for my store, but my lease is for two years. I buy air conditioning, then the landlord raises my rent, and I have to move."

The legitimacy of his fears seems valid, but whether he would install air conditioning if he had a good lease is unknown. Nevertheless, a dozen or so proprietors made comment on the fact that instability of rental conditions precluded any consideration of a large investment in air conditioning.

* * * * * *

The Equipment Developments and Recent Trade Literature columns published each month in HPAC are intended to keep you posted on new and improved products. Be sure to read these departments regularly.

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How to S-p-a-c-e

PIPE SUPPORTS

THE AUTHOR has been involved in the design of several hundred miles of outside pipe lines for large process industry plants. In this work, the economical spacing of pipe supports was of major importance, and a great deal of time was spent in studying the problem.

It was surprising to find that much of the literature on the subject was ultraconservative, and in many cases obvious errors were noticed. As a result of considerable study, a design procedure was developed; and it is reported here for the benefit of others who may have use for it. In addition to developing a theory, a means for quickly applying the theory to everyday problems is presented.

Calculation of Combined Stress Important

Stress in the pipe is the most important criterion in determining the spacing of pipe supports. If it is desired to space the supports as far apart as possible, several stresses in the piping must be figured and properly combined to obtain the total equivalent stress. In the following section, the author is attempting to make a highly refined analysis of stress. In actual practice, many of the stresses calculated may be very small so that they may be neglected; whereas in some cases, all of the stresses must be considered. It is realized that many shortcuts will be apparent to designers for particular cases, but this article is intended to cover general conditions where any one of the several stresses existing in a pipe may be critical. The various stresses that must be calculated are summarized as follows:

- S'', the bending stress caused by the dead weight of the pipe.
- Sw, the bending stress caused by the wind load.
- St, the bending stress caused by thermal expansion of the pipe.
- Sb, the resultant of all of the above bending stresses.
- T, the shear stress caused by twisting of the pipe due to thermal expansion.
- Sa, the axial stress caused by pressure in the pipe.
- Sp, the hoop tension in the pipe wall caused by the pressure in the pipe.
- Sr, the radial stress caused by pressure in the pipe.
- Se, the combined equivalent stress. This stress measures the total effect of all of the above stresses acting together.

This article is principally concerned with the calculation of S'' and Sw. These are bending stresses which are directly affected by the spacing of the pipe supports. The other stresses are mentioned to show how they affect the combined equivalent stress Se. In figuring the bending stress S'', the piping is considered to be either completely restrained at the supports; or to be freely supported, depending on conditions.

The above assumption of fixed or free supports is made in the interest of simplicity. It would be quite impossible to evaluate accurately the degree to which each support is actually restrained. For one thing, a stress pattern is locked in the piping during erection, and this pattern will vary with the method used in erecting the pipe. If a long line of piping were welded together on the ground with temporary supports at frequent intervals, and then hoisted into place and perfectly aligned, the stress pattern would closely approximate theoretical conditions. If, on the other hand, shorter lengths were hoisted into place, and welded together after erection, the stress pattern

THEODORE E. BRIDGE, a mechanical engineer with the University of Chicago in connection with a project under construction for the Argonne National Laboratory, acquired a wide experience in the design of power and of process piping during 15 years with E. I. du Pont de Nemours & Co.; he has also had experience in the field of very high pressures. In his article (of which this is the second part) he presents a rational method for determining the spacing of pipe supports. The data are of particular value in those instances where it is desired to place the supports as far apart as possible. Little information on the matter is available elsewhere

SYMBOLS

C = the term, $(E\ I/144)\ (dy/dx)$ which is proportional to slope.
dy = differential change in deflection, ft.
dx = differential length of beam, ft.
E = Young's modulus of elasticity, psi.
I = moment of inertia of pipe section, in.^{4}
L = distance between fixed supports, ft.
L' = distance between free supports, ft.
M = bending moment, ft-lb.
V = the vertical shear loading on the pipe, lb.
W = the weight of bare empty steel pipe, lb per ft.
X = the distance of any point on the pipe from the origin, ft.
y = the deflection of the pipe at any point X ft from the origin, ft.
Y = a term proportional to deflection = $(E\ I/144)\ y$.

would vary with the location of the final welds. If the final welds could be located at a distance from the support equal to 0.21 L, the stress pattern would closely approximate theoretical conditions.

The possibility of localized stress beyond the yield point, caused by misalignment or by unfavorable erection procedure, is not actually too serious. A factor of safety of four is required by the ASA Code for Pressure Piping. Even if localized stress beyond the yield point is encountered, it will not result in failure because the slight bending of the pipe would immediately relieve the localized bending stress.

The above discussion is intended to show that the assumption of fixed or free supports is justified. It is not considered feasible to estimate the degree to which any of the supports are restrained.

In a Long Line of Supports, Only the End Spans Need Be Considered as Freely Supported. All Other Spans May Be Considered as Fixed—In a long line of supports, only the end span is considered to be freely supported, and the other spans are considered to be fully restrained at the supports. To prove that this is a reasonable assumption, a careful analysis of the bending moment diagram has been made in a piping system (Fig. 4) in which there are four equal spans in the center and one shorter span at either end, making a total of six spans.

By referring to Fig. 4, the following simultaneous equations are readily written from the fundamental beam theory. Take the origin at a and calculate conditions at point b:

$$Mb = Ma + Va\,L - W\,L^2/2 \quad ..[23]$$

$$Cb = Ma\,L + Va\,L^2/2 - W\,L^3/6 \quad .....[24]$$

$$Yb = 0 = Ma\,L^2/2 + Va\,L^3/6 - W\,L^4/24 \quad .....[25]$$

Next, take the origin at b and calculate conditions at point c:

$$Mc = Mb + Vb\,L - W\,L^2/2 \quad ..[26]$$

$$Cc = Cb + Mb\,L + Vb\,L^2/2 - W\,L^3/6 \quad .....[27]$$

$$Yc = 0 = Cb\,L + Mb\,L^2/2 + Vb\,L^3/6 - W\,L^4/24 \quad ..[28]$$

Finally, take the origin at c and calculate conditions at d:

$$Md = 0 = Mc + Vc\,LA - W\,L^2A^2/2 \quad .....[29]$$

$$Yd = 0 = Cc\,L\,A + Mc\,L^2A^2/2 + Vc\,L^3A^3/6 - W\,L^4A^4/24 \quad .....[30]$$

The solution of the above eight simultaneous equations is tedious, but the results may be written as follows:

$$Vc = W\,L\,(1/21 + 2A^2/7 + 5A^3/12)\,/\,(4A/7 + 2A^3/3) \quad .....[31]$$

$$Vb = W\,L(17/28 + 9\,A^2/14) - Vc\,A\,9/7 \quad .....[32]$$

$$Va = W\,L(11 + 6A^2)\,/8 - 1.5\,(Vb + Vc\,A) \quad ...[33]$$

$$Mc = W\,L^2A^2/2 - Vc\,L\,A \quad ...[34]$$

$$Mcd = (W\,L^2A^2/8) + (Mc/2) + (Mc^2/2\,W\,L^2A^2) \quad ....[35]$$

$$Mb = (W\,L^2/2) + Mc - Vb\,L \quad .....[36]$$

$$Ma = (W\,L^2/2) + Mb - Va\,L \quad .....[37]$$

If all spans are designed for the same stress and assuming that the middle spans are fixed and that the two end spans are freely supported, it will follow that: $A = L'/L = \sqrt{2/3} = 0.818$, $A^2 = 0.667$, $A^3 = 0.540$. If these values are substituted in the above, the following equations may be written:

$$Vc = +\,0.506\,W\,L \quad .....[38]$$
$$Vb = +\,0.509\,W\,L \quad .....[39]$$
$$Va = +\,0.491\,W\,L \quad .....[40]$$
$$Mc = -\,0.0806\,W\,L^2 \quad .....[41]$$
$$Mcd = +\,0.048\,W\,L^2 \quad .....[42]$$
$$Mb = -\,0.090\,W\,L^2 \quad .....[43]$$
$$Ma = -\,0.081\,W\,L^2 \quad .....[44]$$

The bending moment for a pipe fully restrained at the supports is expressed by Equation 5, $M = W\,L^2/24 - W\,X^2/2$, given on p. 95 of the October issue. It can be shown that the bending moment will be at a maximum at the supports where $X = L/2$. This may be substituted in Equation 5 to give the maximum bending moment for a fixed span:

$$M = 0.0833\,W\,L^2 \quad .....[45]$$

It will be seen from Equation 43 that the maximum bending moment in the piping system shown in Fig. 4 occurs at support b. Further, it is noted that the bending moment at this point is only 8 percent greater than that calculated for a fixed span in Equation 45. From the above, it is believed that the assumption of fixed supports for the middle spans of a long line of supports is justified provided that the end spans are considered to be freely supported.

The normal factor of safety of four, as required by the Code for Pressure Piping, will easily absorb the slight error introduced by this assumption. This is particularly true when one realizes that the stress is localized at the support. The bending moment reduces very rapidly as one moves away from the supports; and, in the event of unfavorable erection methods, slight bending of the pipe at the support would produce a more favorable stress pattern in the pipe. Moreover, the error introduced by the assumption is probably small when compared to the probable stresses introduced by normal misalignment of the pipe during erection.

All Spans in a Long Line of Supports May Be Considered Fixed If the Piping Is Made to Overhang the End Supports by an Amount Equal to (0.41 L)—Just as in the preceding paragraphs it was proved that the middle spans may be considered fixed if the end spans are considered free, so in the following paragraphs it will be demonstrated that all spans may be considered fixed if the piping is made to overhang the end supports by an amount equal to 0.41 L. In order to prove this point, the piping as shown in Fig. 5 is analyzed in detail.

Take the origin at point a and calculate conditions at point b:

$$Mb = Ma + Va\,L - W\,L^2/2 \quad ..[46]$$

$$Cb = Ma\,L + Va\,L^2/2 - W\,L^3/6 \quad .....[47]$$

$$Yb = 0 = Ma\,L^2/2 + Va\,L^3/6 - W\,L^4/24 \quad .....[48]$$

Take the origin at point b and calculate conditions at point c:

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$$Mc = -WL^2A^2/2 = Mb + Vb\,L - WL^2/2 \quad \ldots\ldots\ldots\ldots [49]$$

$$Yc = 0 = Cb\,L + Mb\,L^2/2 + Vb\,L^2/6 - WL^4/24 \quad \ldots [50]$$

The above simultaneous equations may be solved to obtain the following results:

$$Mc = -WL^2A^2/2 \quad \ldots\ldots\ldots [51]$$
$$Mb = -WL^2(3-4A^2)/28 \quad \ldots [52]$$
$$Ma = -WL^2(1+A^2)/14 \quad \ldots [53]$$

If A is made equal to 0.41, the following values will be obtained:

$$Mc = -0.0835\,WL^2 \quad \ldots\ldots\ldots [54]$$
$$Mb = -0.0833\,WL^2 \quad \ldots\ldots\ldots [55]$$
$$Ma = -0.0835\,WL^2 \quad \ldots\ldots\ldots [56]$$

For a fixed span it has been

TOP TO BOTTOM

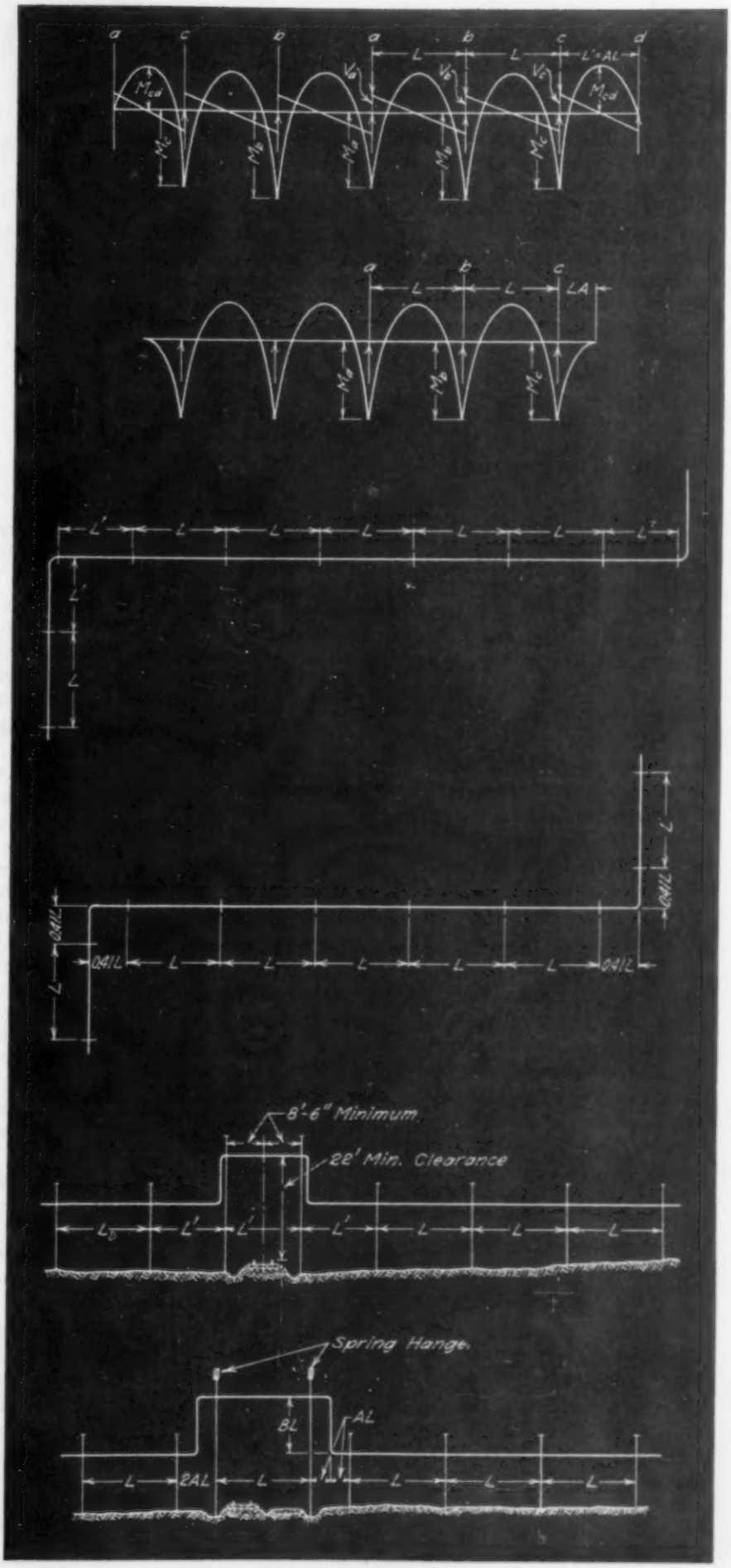

Fig. 4—Bending moment diagram for a continuous beam with seven supports. All spans are of equal length except the end spans which are shorter. The beam is not restrained at the ends

Fig. 5—Bending moment diagram for a continuous beam with five supports. All spans are of equal length, and the beam is made to overhang the end supports to simulate a fixed beam. M = the bending moment at the support indicated by the subscript; L = the spacing of supports

Fig. 6—An actual piping system in which the end spans are considered freely supported, and the other spans are considered fixed. L = the spacing for fixed spans, and L' = the spacing for spans which are considered freely supported

Fig. 7—Showing how all spans may be considered as fixed, if the piping is made to overhang the end supports by an amount equal to 0.41 L

Fig. 8—Normal clearances at a railroad crossing, showing that the spans adjacent to the elbows in a vertical plane are considered to be freely supported due to the flattening effect introduced by the elbow. L = the spacing of supports for spans which are considered fixed; L' = the spacing of supports for spans which are considered to be freely supported

Fig. 9—How all spans at a railroad crossing may be considered fixed, if the piping is made to overhang the supports by a calculated amount which will introduce enough negative bending moment at the support to make it equivalent to a fixed span. L = the spacing of the supports. B is given. The spring hangers may be omitted if the themral expansion in the vertical leg may be neglected

shown, in Equation 45, that $M = -0.0833\,W\,L^2$. From the above, it will be seen that if the piping is made to overhang the end supports by an amount equal to $0.41\,L$, all spans may be considered as fixed.

Still Other Conditions Which Determine Fixed and Free Spans—In Figs. 6 and 7, two systems are shown to indicate fixed and free spans in a long line of supports. L is always used to designate the spacing of supports which are considered fixed, and L' is used to designate the spacing of supports which are considered free.

In Fig. 8 a railroad crossing is shown. Note that flattening effect at the elbows in the vertical offset adds considerable flexibility to the system at that point. This makes the elbow act a little like a hinge so that spans adjacent the elbow must be considered as freely supported.

If the piping at a railroad crossing is made to overhang the supports, as shown in Fig. 9, the spans may be considered fixed. Let A be the ratio of the overhang to the length of the span; that is, $A\,L =$ the length of the overhang. It is desired to choose a value for A which will make the negative bending moment at the support equal to that which would exist in a fixed span:

$$-M = W\,L^2/12 = (W\,B\,A\,L^2 + W\,A^2L^2)/2 \quad \ldots\ldots\ldots [57]$$

Multiply through by $(2/W\,L^2)$:

$$A^2 + A\,B - 1/6 = 0$$

and

$$A = (-B + \sqrt{B^2 + 0.66})/2 \quad \ldots\ldots\ldots [58]$$

The factor B is usually fixed by the dimensions of the piping system, but the factor A may be varied by replacing the supports. If the factor A is calculated by means of Equation 58, the spans adjacent the elbows in the vertical plane may be considered as fixed.

In the foregoing paragraphs, a method of deciding which spans are fixed, and which spans are free, has been developed.

[This is the second of a series of articles by Mr. Bridge on how to space pipe supports. The first was published in the October HPAC.]

Gives Recommendations for Oil Fired Boiler Settings

Many factors affect the design of the setting and furnace of an oil fired boiler. According to Kalman Steiner, fuel engineer, C. Hoffberger Co., the design usually takes cognizance of the engineering problems in the case of industrial and power boilers, but in the case of smaller boilers (under 200 hp) fundamental requirements may be overlooked.

In his paper, Oil Fired Settings for Small Boilers, presented at the recent fall meeting of the American Society of Mechanical Engineers, he reviews the various methods of rating these small boilers and cites the diversity of units in use for expressing boiler capacity. In his opinion, this gives rise to confusion in formulating concepts of relative performance ability of different boilers. By way of illustrating the lack of agreement as to an acceptable standard of the rate of heat release, Mr. Steiner tabulates data from municipal and other regulations governing heat release rates.

Stating that the fundamental basis of boiler setting design should be gross input to the firebox in Btu per hr, the author recommends a basic rate of heat release for design purposes of 24,000 Btu per cu ft per hr. Assuming a combustion efficiency of 70 percent, this corresponds to an allowance of 2 cu ft per developed boiler horsepower. The figure is given as a safe maximum rate for small boilers having a refractory firebox and carrying a moderate, continuous load under natural draft.

Copies of this paper, No. 49-F-16, may be obtained from ASME headquarters, 29 W. 39th St., New York 18.

Trans-Arabian Pipe Line Presents Unusual Problems

The main problem of construction which distinguishes the trans-Arabian oil pipe line from a similar project in this country is due to the remote location and the large supporting organization required to back-up the actual construction staff. Some of the principal design features and a few of the more interesting and unusual problems of construction are described in a paper, Some Features of the Engineering and Design of the Trans-Arabian Pipe Line, by S. P. Johnson, engineer, Standard Oil Co. of California.

In order for the reader to better understand the design of this line, which extends approximately 1070 miles, and to appreciate the peculiar problems involved, the author leads off by reviewing the history of the project. Emphasis is placed on a number of design changes.

Some of the highlights mentioned include the use of two sizes of pipe. Half is 30 in. in diameter and half is 31 in. This permits telescoping to save shipping charges. Originally, it was decided to increase the wall thickness at the high pressure part of the line so that the working pressure could be increased to 885 psi. This reduced the number of required pumping stations from eight to six, a noteworthy saving since each station is, of necessity, an isolated community with recreational, communications, and other facilities not required in this country. More recently the design was altered to permit the addition of intermediate pumping stations to bring the total number to 12. This will mean a maximum capacity of 315,000 bpd when using six stations and over 500,000 bpd when using all stations (300,000 bpd was the capacity originally figured). According to the author, a careful study has indicated that the pipe line will be under no more net stress if no attempt is made to permit expansion and contraction and it is simply held rigidly in place, being suitably anchored at changes in direction. Many other design and construction factors are cited.

The paper was recently presented at the semi-annual meeting of the American Society of Mechanical Engineers, 29 W. 39th St., New York 18.

Graphical Symbols for Use on Drawings—The American Society of Mechanical Engineers, 29 W. 39th St., New York 18, has published several graphical symbol standards developed under the procedure of the American Standards Association. Included are Graphical Symbols for Welding (Including Instructions for Their Use) No. Z32.2.1-1949—50 cents; Graphical Symbols for Pipe Fittings, Valves, and Piping, No. Z32.2.3-1949—40 cents; and Graphical Symbols for Heating, Ventilating, and Air Conditioning, No. Z32.2.4-1949—40 cents.

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Modern Thinking Applied to

DUCT DESIGN

KIRBY WALKER draws upon his experience both in the heating and ventilating field and in wartime research and engineering to discuss duct design. A licensed professional engineer, he was engaged on heat transfer and combustion projects for two large aircraft companies during the war. He has had a wide experience in the heating and ventilating industry, and has supervised manufacturers' laboratory and development personnel

AN INCREASE in area in a duct reduces the air velocity and if not handled very carefully can result in very high friction because of the eddy currents produced by the high velocity stream entering the larger space where the velocity is lower. Increases in duct area cause much higher losses than area reductions.

Practically speaking, when there isn't a sharp edge to cause a vena contracta, the largest loss in enlargement occurs when an opening discharges into a plenum. (By "plenum" is meant a space where the velocity is effectively zero.)

The stream entering the space of lower velocity can have a vena contracta induced by a sharp edge. When this occurs, f' may be greater than 1.00. On the other hand the stream may come out of the opening in substantially straight lines as it would leaving the end of a straight pipe. Under these conditions the loss is substantially f' equal to or less than one. Such a loss is shown by the 180 deg line in Fig. 12.

The diverging pipe is important in duct work. Data available on enlargements are contradictory and little information has been assembled and plotted f' vs. *R.N.* as it should be. Because the available data are sparse and unclear, Fig. 12 is presented despite its extreme inaccuracy. It has been drawn roughly to cover rectangular sections in which f' is possibly one-third larger than f' for round sections; that is, cones. Fig. 12 is for enlargements with pipe attached. Without attached pipe, losses are larger than shown.

SYMBOLS

A = (area) = (square feet); usually duct cross section

D = (equivalent diameter) = (feet) = $\frac{1}{3} \frac{\text{(sectional area, square inches)}}{\text{(perimeter, inches)}}$

f = (friction coefficient, straight pipe) = $h\ D/L\ q$; plotted vs. *R.N.*

f' = (friction coefficient, shock losses) = h/q

h = (loss of total pressure) = (inches 32 F water column)

$h = q f'$; shock losses, obstructions

L = (length of duct) = (feet)

q = (one velocity head of the average velocity) = (inches, 32 F water column)

R.N. = Reynolds number

Enlargements Made at Elbows

In the case of enlargements made at elbows, use the f' for the particular type of ell, curve radius, etc., increasing it somewhat to allow for the effect shown in Fig. 12. The resulting f' is used with q upstream in the small area. Limit this f' to one.

The amount to add to allow for enlargement can be estimated by laying out the upstream and downstream sections a distance equal to the centerline of the ell. Estimate the largest angle and refer to Fig. 12. Add the "enlargement f'" to the "elbow f'" and use one or the total, whichever is least.

An enlargement at an ell is just as important as an enlargement anywhere else. A contraction at an ell doesn't matter. It can't give much loss.

Enlargement Distribution

Besides being a source of loss, an enlargement may give bad distribution downstream. Roughly speaking, there is an included angle of 14 deg that may be counted on. If distribution is required through a greater included angle, vanes or some form of guides must be resorted to—just as is done to get distribution and lower drop through an elbow. Fig. 13 illustrates this general rule.

We hear of the relative merits of "pushing" or "pulling" air through heaters and other apparatus. Actually it doesn't matter (except for leakage) whether you pull or push. It is the 14 deg jet that is objected to or the stratification at a fan outlet. Fig. 13B would have the jet effect shown regardless of whether the air flow was caused by "pushing" or "pulling."

Flush Takeoffs

There are certain types of apparatus supplied from ducts that require a uniform distribution of air across the duct feeding them; particularly, certain types of ceiling outlets and fan inlets. A fan's rating, for instance, is seriously affected if air enters its inlet poorly distributed. Consequently, there has grown up a practice of taking such an outlet out of the side of a duct with no attempt to build elbow principles into the connection. Since the connection is used to get good distribution, it is important to make the takeoff smaller than the

supply duct all around. Such connections are shown in Fig. 14.

Notice that in turning fluid at right angles with no attempt at streamlining, approximately one velocity head is lost. The sharp entrance may cause a loss of ½q in the takeoff duct, so the total loss of such a connection is great. In the case of a fan inlet, this loss can be cut considerably by using a properly vaned elbow or a good straight duct leading up to the fan and then carefully transforming from a rectangular duct to a round duct the same size as the fan entrance.

Flush Takeoff at Fan Inlet

To illustrate the point made regarding fan inlets, Fig. 15 shows a fan connected in two ways; first, by a well designed elbow and transformation, and then by a flush takeoff. Both designs would give reasonably good flow distribution at the fan inlet.

For the fan connection using the flush takeoff:

10-11	$h = 0.0625$
11-12	$h = 0.0800$
	0.1425

For the elbow and transformation:

4-5	$h = 0.0188$
5-6	$h = 0.0011$
	0.0199

We thus see that the well designed fan inlet can cause but one-seventh the friction of the poorly thought-out boxlike flush takeoff connection.

Transformations

Transformations may be used to increase or decrease duct area, to change from one set of dimensions to another set, to shift a duct position or direction without the use of an elbow, or to accomplish some combination of these functions. For most purposes rectangular flat side construction may be used. Sometimes dimensions and area may be changed by use of an ell. If there is room for a full radius and the aspect width is 0.8 or over, a simple radius ell may be used without vanes.

In general, reductions of area do not produce losses as large as increases of area. Where you pass from one shape to another by the use of flat planes you can at some intermediate point pass through an area very different from the areas

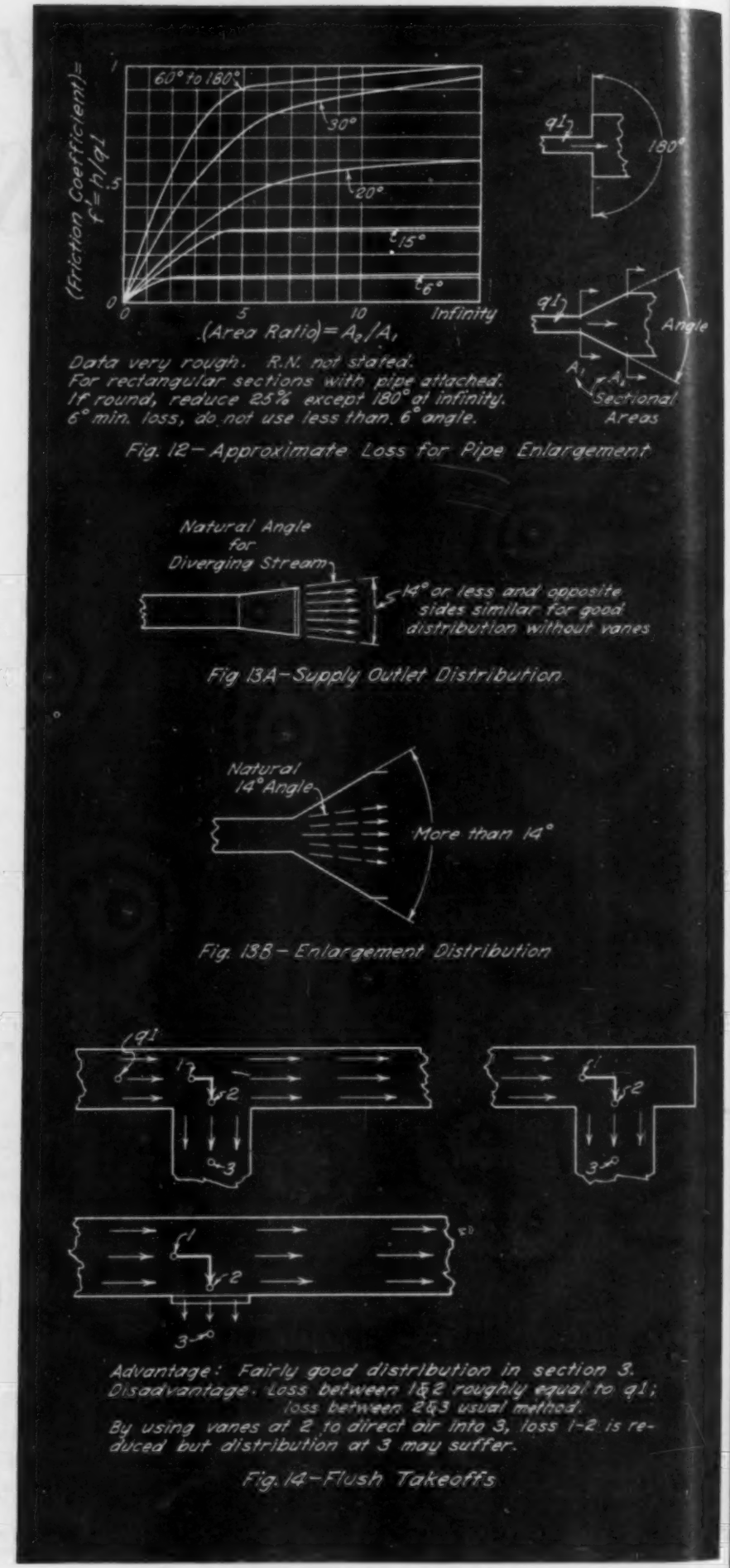

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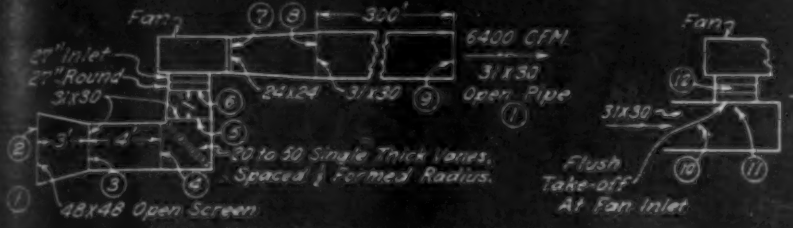
Fig. 15



Obstruction		Major Dim.			R.N.		f or f'	q for f or f'		Loss of Total Press. h=qf or h=qfL/D
Loc.	Description	Loc.	FPM	D	Loc	Value		Loc.	W.C.,in.	
1-2	Proj. Pipe Entrance	2	400	4	2	168,000	1.	2	0.01	=0.0100
2-3	15° Reduction, Same as 3'-Piece of 3-4				3			3		=0.0013
3-4	4' Str. Pipe Fig. 2	3-4	1000	2.5	3-4	260,000	0.017	3-4	0.0625	=0.0017
4-5	Square Turn S.T. Vanes Fig. 9	4	1000		*		0.3	4	0.0625	=0.0188
5-6	15° Transformation Same as 1' Str. Pipe	6	1600	2.25	6	378,000	0.016	6	0.16	=0.0011
6-7	Fan Adds to T.P.									
7-8	6° Enlargmnt. Fig. 12	7	1600	2	7	336,000	0.1	7	0.16	=0.0160
8-9	300' Str. Pipe Fig. 2	8-9	1000	2.5	8-9	260,000	0.017	8-9	0.0625	=0.1280
9-10	Open End	9	1000	2.5	9	260,000	1.	9	0.0625	=0.0625
10-11	Rt Angle Turn Fig. 14	10	1000	2.5	10	260,000	1.	10	0.0625	=0.0625
11-12	Sharp Flush Entrance	12	1600	2.25	12	378,000	0.5	12	0.16	=0.0800

*For R.N. use opening between vanes to figure equivalent diameter.

Fig. 16A

Fig. 16B

C, Converging Angle
S, Separation Angle

Fig. 16C

Case	Area Downstream from Joint	Largest Separation Angle	Converging Angles	Loss: Take q at Joint
1	Constant or Decreasing	7° or Less	Ignore	Straight pipe: Use mean area.
2	Constant or Decreasing	Over 7° & All-around Perimeter	Ignore	f' is 0 for 7° f' is 0.25 for 45° f' is 0.50 for 90°
3	Increasing	Any	Ignore	Double largest separation angle; enter Fig. 12; use A_2/A_1 to get f'
4	Increasing Constant or Decreasing	None	All	Zero
5	Constant or Decreasing	Over 7° & Only Part Way Round	Ignore	f' is 0 for 7° f' is 0.5 for 45° f' is 1 for 90°

Above a rough guess - See text.

at either end of the transition. In such a case it is sometimes well to break the piece into two or more transitions. If you will scale section *A-A* in Fig. 16B you will see that the area at *A-A* is 4, and that the area at *1* and *2*, is 3.

Since a transition can take an infinite number of shapes, it is necessary that we have a criterion for judging the possible losses in such a fitting. Reliable data are not at hand but the slope of a break in duct shape can give us some ideas regarding a particular case. Accordingly, Fig. 16 illustrates a converging angle and a separation angle where a duct face departs from a straight run. The converging angle may be ignored. It does not tend to cause turbulence and loss. The separation angle indicates a loss. Where such an angle exists the fluid tends to break away to separate from the duct wall and dissipate energy as turbulence.

A separation angle can occur at the first and second junctions of a transformation so there are two places where loss occurs and these must be added to get total loss for the transformation. These junctions are numbered *1* and *2* in Fig. 16.

Since there are no definite test data plotted vs. *R.N.* for transformations, the tabulation in Fig. 16C is presented as a rough approximation. Fig. 16A could vary from straight pipe to two 90 deg square turn ells if there were a large enough offset. If it became two ells, the separation angle at each joint would be 90 deg part way round, Case No. 5, Fig. 16C, and total for both joints would be $h = 2q$.

If a transformation were converging all around and this were made to approach a sudden reduction in area, Joints *1* and *2* would be in the same plane; Joint *1* friction would be zero (Case No. 4) and Joint *2* would have a 90 deg separation angle all around (Case No. 2) and the loss would be $h = 0.5q$; q being taken in the downstream pipe, of course. This answer would be reasonable for a sharp entrance —which is what it would have become.

The reader should understand that Fig. 16C is just a rough guess.

[This is the fifth of a series on this subject by Mr. Walker.]

SAM LEWIS' PAGE

How to Connect Two Boilers With Different Water Lines?

In my September "Page," I told a tale about two steam boilers trying to work together when the water level in one boiler was $18\frac{5}{8}$ in. higher than that in the other. A reader has sent in a solution which he thinks would make it unnecessary to lower the steel boiler, elevate the cast iron boiler, install a pump, or trade one of the boilers for a mate to the other.

The proposed solution is suggested in the accompanying drawing. Each boiler is to have a Hartford loop, but the loop on the higher boiler is peculiar in that the tee in the loop, into which condensate returns, is below the waterline, and that the vertical pipe that brings water to this tee extends upward to an elbow, the bottom of which is at the waterline of the higher boiler. Water then returns to the higher boiler until, having filled it to the waterline, the water flows to the lower boiler through a standard Hartford loop wherein the tee in the loop is at, or slightly below, the waterline.

The normal wet return connections near the boiler room floor are installed with valves marked *1* and *2* on the drawing, permitting closure of condensate flow to each boiler.

A second wet return some little distance above the normal return connecting the loop tee at the water level of the higher boiler to the return going to the loop of the lower boiler is suggested, with a valve *3* in this pipe.

The plan is to close valve *2* when both boilers are being fired, and to open valve *3*. The condensate then is invited to flow through valve *1* and to fill the higher boiler to its waterline; after which the condensate can flow through valve *3* to fill the lower boiler.

If the lower boiler is to be fired alone, then valve *1* is to be closed, valve *2* is to be opened and valve *3* is to be closed. Similarly, to operate the higher boiler, valve *1* is to be opened and valves *2* and *3* are to be closed. This, of course, presumes that the appropriate steam header valves will be opened or closed.

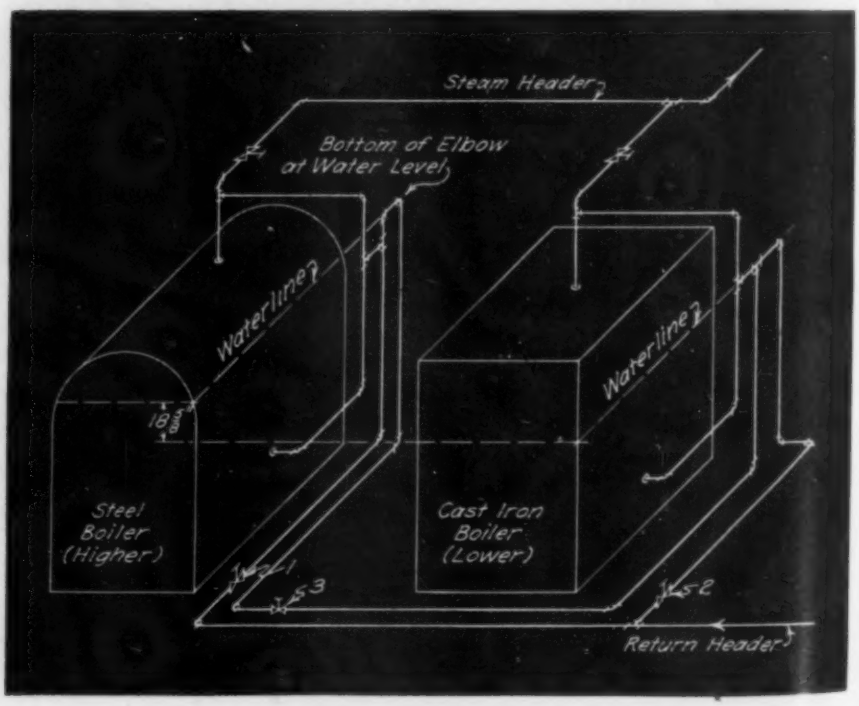
I was asked to criticise the solution and here is my answer to it:

Granted that the desire is to fire both boilers as a team, then both boilers must receive enough water to maintain the required water level in each.

Since the steam header valves are open, the pressure above the water will be equal in both boilers. With valve *2* closed, as suggested, no water can enter the lower boiler until the water level in the higher boiler has been reached. Such a level requires a water pressure in the return mains of sufficient intensity to reach the level of the higher boiler. This pressure must cause the water to flow through the valve *3* into the lower boiler and the level of the water in the higher boiler cannot rise higher until flow through valve *3* ceases. Flow through valve *3* cannot possibly cease until the pressure on each side of the valve equalizes and that can only occur when the lower boiler water level rises $18\frac{5}{8}$ in. above the desired level. Thus the lower boiler will have no steam space whatever and will cease to be a steam boiler, and the scheme will fail.

Consider a U-Tube

Consider a U-shaped glass tube partly filled with water. If the air pressure against the surface of the water in each vertical leg of the tube is equal, the water level in each leg will be the same.



Scheme proposed by a reader for connecting two boilers with different waterlines. How the valves would be operated is described in the text. Mr. Lewis explains why this method will not be satisfactory.

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Even if the U-tube below the water level is as crooked as a happy pig's tail, the water in each leg will equalize. Feeding the lower boiler through the higher one will not in itself prevent flooding the lower boiler.

Each ounce of pressure that might bear down on the water in one of the legs of a U-tube will make a difference of 1.73 in. between the two water surfaces.

In this case of the two boilers we really have a U-tube and each boiler is a leg of a U-tube.

All of us have noted differences in water level between two adjacent steam boilers that are duplicates of each other and that are in free communication via the steam header and wet return header. A difference in steaming rate, or a partly closed steam valve, or a difference in the number or relation of fittings and valves in the respective steam connections to the header, can account for pressure differences on the water in each boiler that will be manifest on the gage glasses. There is a difference in desired water head for these two steam boilers of 18⅝ in., or 18.62/1.73, which is about 11 oz. As long as this pressure difference is not compensated for, we can be sure that the lower boiler will become a water heater whenever it is tried to operate both boilers simultaneously without positive separate water level controls—such as by individual waterline governors and a pump.

I hope that the reader who proposed this solution will believe me, or will try the scheme on his next similar job and then report the results.

INFORMAL comment on heating, piping, and air conditioning matters is given in this regular feature by Samuel R. Lewis, consulting mechanical engineer, member HPAC's board of consulting and contributing editors

Heating Cable Guards Piping Against Freezing

Twelve years of service has proved the usefulness of electric heating cable in the Cleveland union terminal and has substantially reduced the expense of maintenance during freezing weather, according to engineers at the terminal as quoted by General Electric Co.

The cable was installed on 88 water service laterals in the terminal, coach yard, and station. Prior to 1937, considerable trouble was experienced with the water laterals in the wintertime. Freezing and bursting pipes were a routine maintenance problem, and station and coach yard platforms had to be dug up frequently to repair damaged pipes.

In 1936 the terminal began its program of installing the heating cables on these laterals. The cable used was No. 19 nichrome resistant wire, insulated with asbestos, varnished with cambric, and covered with lead sheath. Each circuit consisted of 60 ft of heating cable connected to a 110 volt direct current circuit.

The 60 ft circuit was divided into 12 to 15 ft units, each coiled around the water lateral pipe to be protected. The intermediate connections between the heating units were made through buried, nonmetallic, trench type cable. The temperature of this heating cable in open air, when connected to the proper voltage, is about 165 F.

Pipe laterals were made accessible for the cable installation and maintenance by constructing small brick chambers with removable metal covers.

Recently the terminal made additional use of heating cable. Compressed air is supplied to station and coach yard tracks from a central air compressor plant. At this plant an outdoor type, air cooled aftercooler is used to remove moisture from the compressed air. During freezing weather trouble was experienced draining off trapped moisture. Sixty feet of heating cable was wrapped around the drainpipes and traps which were then covered with waterproof insulation.

By applying heat to the pipes and traps during subfreezing weather, the terminal engineers are able to guarantee troublefree operation of the aftercooler at all times.

COMPETITION SPURS CLEANING OF COAL

The growing need for cleaning coal, already a tremendous influence on the national economy, is being spurred by the rapid depletion of the purer, thicker coal beds of the east, the American Society of Mechanical Engineers was told at its fall meeting. Cleaning coal reduces the equipment necessary to deliver it to market, increases the efficiency of every coal burning unit and increases byproduct return, said Thomas F. Downing, Jr., of the Philadelphia Electric Co. He defined cleaning coal as the practical reduction and retention at the mines of materials having a relatively low combustible content.

Cleaner coal is needed to compete with other fuels and as the cleaner seams of producing areas are exhausted, inventors and manufacturers are trying to keep pace with the growing need for cleaning the "black diamond," Mr. Downing said. The rapid growth of mechanical loading in recent years has reduced operating costs for mine operators, he said, but presents a serious setback to clean coal.

CONSTRUCTION A BRIGHT SPOT

Construction should continue as one of the brighter segments of the national economy through 1950, James M. Ashley, president of the Producers' Council, national organization of building products manufacturers, said in a report to the council's board of directors at the organization's annual meeting. Mr. Ashley said that producers of materials and equipment are prepared to meet any demand which conceivably might arise in the future and added that the supply of skilled labor also should be entirely ample.

"Optimism for the future is based on the fact that there is a considerable volume of work in all categories of building awaiting more favorable investment conditions," he stated.

QUESTION OF THE MONTH

Suggests a Way to Re-Use Water from Cooling Coils

WATER FURNISHED by a deep well pump supplies 75 gpm to a set of coils for air conditioning a factory area, and the plant engineer wants to use the water after it has left the coils. However the factory demand for water fluctuates anywhere from 10 to 50 gpm. Using the present setup, what is the best way to obtain constant pressure for plant use, constant pressure for the air conditioning coils, and adequate excess water disposal?, asked "G.M." in a question published in the September issue.

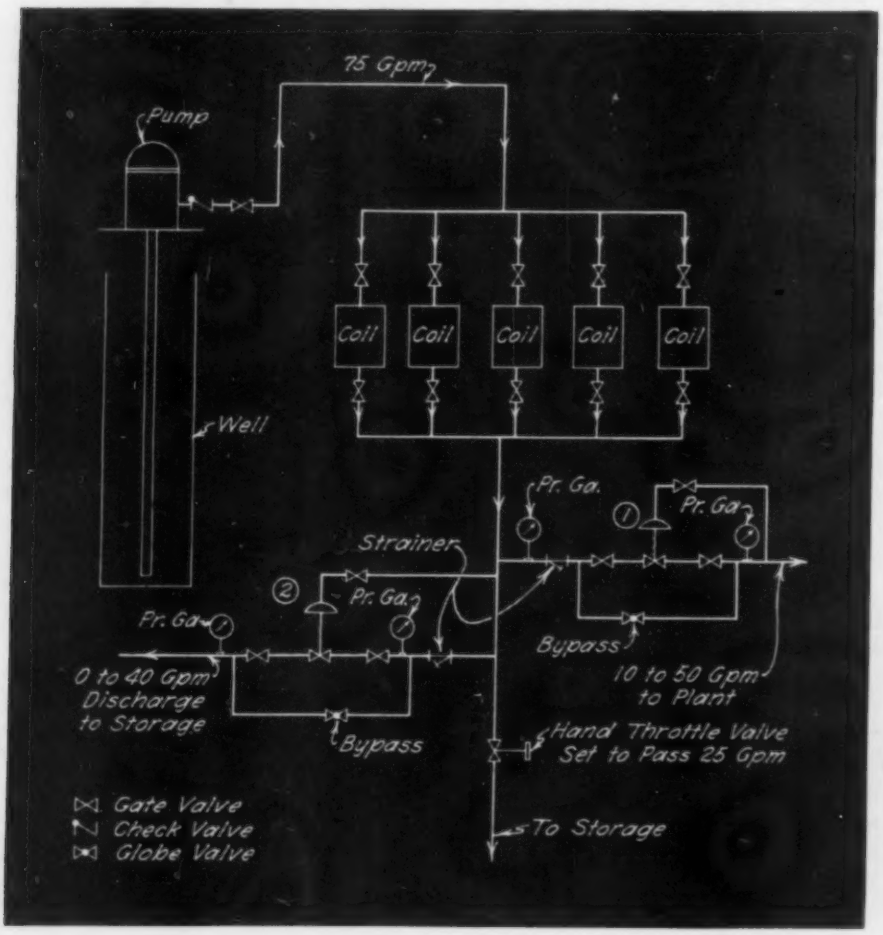
I suggest that "G.M." provide two diaphragm motor valves, one set to maintain a constant pressure on the downstream side of the valve on the supply line to the plant. This valve would have a maximum capacity of 50 gpm at the required operating pressure range and a minimum capacity of 10 gpm. The other valve would be placed on a branch line which would take off the same header as valve No. 1, as shown in my sketch. This would be the same type of valve but controlled by the pressure in the line upstream of the valve. It should have a capacity of 40 gpm maximum down to a minimum of zero, at the desired operating pressure. Its control would be set so that when valve No. 1 was delivering only 10 gpm to the plant and the pressure started to build up on the upstream side of valve No. 1, then valve No. 2 would open and discharge enough liquid to take care of the excess.

The discharge from valve No. 2 should be run to large storage tanks and saved for some other use rather than wasted.

Another line should take off on the downstream side of the coils and the two control valves that would be capable of discharging 25 gpm constantly, with a hand throttling type of valve. This would provide for the disposal of the excess water not otherwise taken care of.

For precise operation the valves should be equipped with air operated pilot controls on the motor valves. All motor valves should be provided with bypass arrangements so that they may be removed from service temporarily for inspection and repair.

An air filter should be placed in the air line ahead of the air pilot control.—S.H.



This sketch is submitted by "S. H." to show a method for re-using water from a set of air conditioning coils

READER WANTS DATA ON OFFICE CONDITIONING

You are invited to answer the following question from a reader of HPAC. Suitable comment on the matter will be welcomed and—space permitting—will be published and paid for at regular rates. Sketches and photos are particularly desired. Address the Editor, HEATING, PIPING & AIR CONDITIONING, *6 N. Michigan Ave., Chicago 2.*

OUR COMPANY is a subscriber to HPAC and we thought you might possibly be in position to give us some help on a problem which we have in connection with air conditioning for our main office building.

Our building is approximately 60 x 60 ft inside dimensions, with two main floors and one-half of the third floor developed for working space, and a full basement, which probably will be developed for working space at a later date.

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We do a contracting business which means, as far as this office is concerned, the people are doing estimating—that is, working with plans and specifications, and other paper work, as well as routine clerical work, etc. We have about 60 regular employees in this office.

I am of the opinion that an air conditioned building would result in a large increase in efficiency of the office personnel, as well as improved health conditions due to the stable temperature and humidity conditions under which the people would be working. However, some of our people do not go for the idea, their objections being based mainly on a theory that air conditioning is not conducive to good health for the employees.

YOU ARE invited to submit a heating, piping, or air conditioning question for publication here. You are also invited to submit answers to previously-published questions from others. Write the Editor, Heating, Piping & Air Conditioning, 6 N. Michigan Ave., Chicago 2, Ill.

It occurred to us that you very likely have in your files, or could refer us to, certain pamphlets or literature by recognized authorities on the subject that the overall efficiency and health conditions of people working in an air conditioned building would be far superior to the same group working in a building that is not air conditioned.

It seems to me that I have seen literature on the subject—written by those who are supposed to know—to the effect that general working efficiency of the working personnel is X percent better, and that the general health conditions of these people is much better, with air conditioning.—M.A.

[Editor's note: Some information—including articles which have appeared in HPAC—has already been sent "M.A." in answer to his request. However, there must be a great amount of unpublished data available which would aid him.]

Gives Practical Pointers On Operation, Maintenance

Maintenance and operation of a large air conditioning or heating system are simplified and improved by numerous design and installation details, as well as by careful study and analysis of what should be done when. This was brought out in a recent talk by W. A. Stahl, operating manager of the Merchandise Mart, big Chicago building, before the Illinois ASHVE chapter.

Housings in fan rooms—for example—should have good lighting that can be switched on from the outside, he said, for if an extension cord is required the mechanic is often inclined to skip an inspection. Remote control for all fans—the Mart has about 250—is being considered to improve operation. However, with manual control, each fan must be visited at least twice a day, which is an advantage from the inspection standpoint.

As an example of the importance of analysis of maintenance work, Mr. Stahl mentioned difficulty in one air conditioned space from excessive dirt on the ceilings around the air diffusers. It was found that in this space, the floors were swept in the evenings when the fans were on and the aspiration of room air was the principal cause of the streaked ceilings. At least one large building is considering the use of plastic rings around ceiling air diffusers to aid cleaning. Use of ceiling diffusers makes tenants more conscious of dirt in the air.

More attention should be given to keeping dirt out of buildings, and any methods of keeping dirt from being tracked in are worthy of consideration. Snow melting sidewalks might aid in this way, said Mr. Stahl; he also mentioned his interest in a battery driven scrubbing machine for lobbies during the day, when an electric cord could not be used to furnish power to such a machine.

Dirt from the duct system at the start of the heating season each year is also a problem requiring attention, and humidification is now being used at the Mart even in the straight ventilation systems to minimize this difficulty.

As to complaints from occupants on conditions maintained in the spaces—always an operating problem in any building—Mr. Stahl said that it is important to have all complaints from any one tenant's space clear through one responsible person. It's impossible to please all occupants of large space, and if the complaints don't clear through one person, the operating force will wind up by pleasing no one, he felt.

The Mart, due to its location, is able to use river water for condensing purposes, the water of course being returned direct to the river. This water is well-loaded with silt, so the tubes of the centrifugal refrigeration machines are cleaned with special nylon brushes shot through them with a good slug of water under pressure. This method is doing a good job on the tubes.

Particularly with today's labor prices, good tools are a good investment and it pays to keep your men supplied with the right tools—lathes, welders, shears, brakes, electric saws and drills, pneumatic hammers, etc. Impact wrenches for example, have been found to be great timesavers on cover plates, etc., at the Mart.

Some trouble is beginning to be experienced with hot water service piping at the Mart, now that the building is over 20 years old, Mr. Stahl said—most of it being on the upper floors where the pressure is lower. This is attributed to the higher release of oxygen with the lower pressure. Corrective steps include reduction of water temperature to 130 deg, with booster heaters for such tenants as restaurants, and the use of circulating pumps to get the water around faster. Water treatment is also widely employed.

How careful analysis can reduce the cost of a maintenance operation is exemplified by the cost of changing electric lamp bulbs, Mr. Stahl remarked. Lamps were formerly changed when they burned out, and the cost of changing a bulb was 30c. They are now changed according to a group replacement plan, all those in a group being replaced when they have been in service for an estimated 80 percent of their life. This has reduced the cost of changing to 10c per lamp instead of 30c, he said.

Greyhound "Scenicruiser" Features an Unusual Air Conditioning System

GREYHOUND LINES' new "Scenicruiser," seating 43 passengers, is designed for new luxury in intercity bus travel. Construction is such that increased visibility is provided from every seat. A striking feature is a rear lounge compartment with eight seats grouped in a semi-circle facing the broad, curved rear observation windows. At the driver's command are 2-way communication radio, a public address system, and a 3 hr musical program on a wire recorder.

The unusual air conditioning system utilizes the structural parts of the body framing and walls as air ducts for radiant heating and cooling. The ventilating system is capable of drawing in outdoor air at a rate sufficient to change the air every 40 sec. This air is cooled and dehumidified or heated as required and circulated through vents at the window ledges.

Temperatures specified are 70 F during the day and 74 F at night when heating, and 75 F and 79 F when cooling. The coach temperature can be varied from the driver's location within plus or minus 3 deg.

The heating equipment used is a gasoline fired, liquid medium heater of approximately 150,000 Btu capacity connected to a heater coil mounted in the plenum chamber. The refrigeration unit is of approximately 7½ ton capacity and consists of a three cylinder "Freon" compressor directly connected to a 23 hp gasoline engine. This engine also drives the cooling and scavenging blowers for the condenser and the engine compartment, and the electrical alternator. The evaporator is mounted together with the heater coil in the plenum chamber so that the air passes first the heater coil and then the evaporator.

All of this equipment is located roughly at the middle of the coach below the seat floor. From the plenum chamber, air ducts are carried forward and aft to distribute the air into the floor and side framing of the coach. The side framing members have perforated webs to let the air pass from inside the floor structure up to the air outlets which are located just below the window capping. This method of distributing the air is used throughout the coach and it provides a curtain of conditioned air rising across the window so as to prevent the intense chill or heat felt by the passenger adjacent to the window in winter or summer.

Recirculated air grilles are located at the rear lounge and in the step risers between the fore and aft decks. Some of the recirculated air is passed over the two controlling thermostats which actuate the cycling operations of the system. Air exhausts are located in the toilet, at the front end of the bus, and in the rear of the lounge. The last two exhausts operate on overpressure in the coach.

The thermostats are of the mercury tube type. The controls are electro-hydraulic where electrical actuation and hydraulic operation is coordinated. The system has its own hydraulic pump unit electrically controlled and driven.

The operation of the air conditioning system is based on a differential between the two thermostats which actually senses the approach to control temperature, be it on heating or cooling, so that the equipment can be actuated early enough to prevent overshooting the control temperature by an uncomfortable amount.

The elevated night temperature feature is automatically actuated when the driver turns on the headlights. The reason for this feature is passenger comfort during sleep when frequently a chill is experienced at normally comfortable day temperature.

The "Scenicruiser" is now making test runs.

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Heating, Piping & Air Conditioning

AMERICAN SOCIETY of HEATING and VENTILATING ENGINEERS

Journal Section

Removal of Internal Radiation by Cooling Panels

By Merl Baker*, Lexington, Ky.

SUMMARY—Internal radiation emitted from electrical lighting filaments may be efficiently removed from a conditioned space by cooling panels. Because of the high temperature of incandescent filaments, the receiving panel or *sink* may be operated at a relatively high temperature which may be obtained by use of unchilled water. Fortunately, for most light colored painted surfaces, the visible portion of the radiation spectrum is highly reflected while the infrared, constituting the bulk of filament radiation, is extensively absorbed.

The results of research conducted by the author show that a maximum of approximately 65 percent of the electrical input to a large incandescent filament may be removed directly from the conditioned space. Specially designed reflectors may be used to concentrate the radiation on particular surfaces for removal.

INTERNAL energy sources within a structure, both radiant and convective, may represent a large portion of the total cooling load. Any body within the cooled enclosure at a surface temperature greater than the ambient air temperature is a convective heat source, while any body within the enclosure at a surface temperature greater than the mean radiant temperature of the enclosure surfaces is a radiant source. Therefore, a body may be a convective source, radiant source, or simultaneously, a convective and radiant source depending solely on the temperature relationships.

The rate of energy dissipation by natural convection from a vertical, or horizontal surface facing down, varies as the 1.12† power of the temperature difference between the surface and ambient air. In contrast, the dissipation rate by radiation is proportional to the difference of the fourth powers of the absolute temperatures of the emitting surface and mean radiant temperature of the enclosure. Therefore, for a simultaneous radiant and convective source, the most general type, the radiation component becomes of far greater significance as the surface temperature increases. For high temperature sources such as incandescent filaments, the radiation component reaches a magnitude in the order of 90 percent of the total energy dissipation.

An analysis of the mechanisms of heat removal from an enclosure reveals that the radiant and convective components of internal energy sources must be treated separately. The convective component leaves the emitting body and raises the temperature of the surrounding air after overcoming the resistance of the surface film. The action of the radiant component is much more direct in that it is transmitted directly from the surface of the emitting body to the enclosure surfaces.

Panel Performance

Temperature. Any surface which is maintained at a temperature lower than other bodies within a system becomes effectively a cooling panel, and this surface acts as a *sink* for radiation striking it. The panel may receive energy directly from a source, indirectly by reflection of source emission from other surfaces, or indirectly by reradiation from other surfaces. The source described previously is not limited to any particular type, but the nature of the source will control the temperature required at the panel surface for effective removal. For an enclosure with all surfaces maintained between 50 and 70 F, the loss by radiation and convection from human occupants will vary approximately as the linear function of the temperature differential. In contrast to operation with low temperature sources, the temperature of the panel becomes nearly insignificant when high temperature sources are involved. The radiant energy received by a 70 F panel is insignificantly different from that received by a 50 deg surface. The possibility of panels operating at relatively high temperatures in removing a lighting load is an outstanding economic factor. For installations in which cooled air is circu-

*Assistant Professor of Mechanical Engineering, University of Kentucky. Junior member of ASHVE.

†Panel Heating and Cooling Analysis, by B. F. Raber and F. W. Hutchinson. (John Wiley & Sons, Inc., New York, 1947.)

lated to produce comfort, uncooled water may be used to remove a fraction of the radiant component of the lighting load, or about 70 percent of the electrical input to the filament. For rooms possessing external exposures, the removal of the lighting load is largely effected by the cooling action of the outside walls.

Absorptivity. The absorptivity of all nonmetallic surfaces for long wavelength radiation is generally between 0.9 and 0.98. Therefore a simplified relationship neglecting multiple reflections may be applied to nonmetallic receivers for low temperature radiation with an approximate error not exceeding 4 percent. This fortunate simplification is not valid however, when the radiation is at short wavelengths, near infrared and shorter, as the absorptivity of some nonmetallic surfaces may become as low as 0.2 for visible radiation. Therefore it is necessary to consider multiple reflections in analyzing the transmission of energy emitted from high temperature sources. The basic radiation equation may be applied if the emissivity factor is known. Since this determination is extremely complex for the case of an enclosure with multiple reflections, it is convenient to make some simplifications. By considering the transfer of radiant energy initially emitted from one equivalent gray area A_1 to another gray area A_2 when the surrounding surface consists of one gray area A_3 which may *see* itself, and when all other surfaces are black, the exact equation† for the emissivity factor for transfer between area A_1 and A_2 is given by

$$S_{1\text{-}2} = F_2F_3F_4\ [F_{1\text{-}2} + (1-\alpha_3)\ F_{1\text{-}3}F_{3\text{-}2}F_1]\ e_1\ \alpha_2 \quad \ldots\ldots\ldots (1)$$

where

$$F_1 = \frac{1}{1 - F_{3\text{-}3}\ (1-\alpha_3)} \quad \ldots\ldots\ldots (1a)$$

$$F_2 = \frac{1}{1 - (1-\alpha_1)\ (1-\alpha_3)\ F_{3\text{-}1}F_{1\text{-}3}F_1} \quad \ldots\ldots\ldots (1b)$$

$$F_3 = \frac{1}{1 - (1-\alpha_3)\ (1-\alpha_3)\ F_{3\text{-}3}F_{3\text{-}3}F_1} \quad \ldots\ldots\ldots (1c)$$

†Panel Heating and Cooling Analysis, by B. F. Raber and F. W. Hutchinson. (John Wiley & Sons, Inc., New York, 1947; Chapman and Hall, London, 1947.)

Symbols

a = absorptivity
A = area, square feet
C = conductance of wall, inside surface to outside air, Btu per (hour) (square foot) (Fahrenheit degree)
E = strength of source, Btu per hour
E' = $m\ E$
m = fraction of source dissipation that reaches the surfaces
E_n'' = fraction of energy reaching surface n by radiation from filament
F = configuration factor for radiant heat exchange between surfaces. Subscripts of F refer to the surfaces by number or by letter. e.g. subscript (1-2) refers to exchange from surface 1 to surface 2.
h_c = unit conductance for thermal convection Btu per (hour) (square foot) (Fahrenheit degree).
h_r = unit transfer for thermal radiation, Btu per (hour) (square foot panel) (degree difference of MRT and panel)
L = subscript referring to filament
(MRT) = mean radiant temperature, Fahrenheit
n = any surface
Q = quantity of heat transfer, Btu per hour
$S_{1\text{-}2}$ = shape factor allowing for multiple reflection in evaluating the radiant transfer from surface 1 to 2
T = absolute temperature, Fahrenheit
t = temperature, Fahrenheit (subscripts: 1,2,3, inside surface wall No. 1,2,3, a = inside air, o = outside air)
W = rate of air exchange (ventilating rate), pounds per hour
x = subscript referring to combination of all surfaces excluding n

Subscripts

1,2,3, etc. refer to numbered walls.
b,c,p,r refer to back of panel, convection, panel and radiation respectively.

$$F_4 = \frac{1}{1 - (1-\alpha_1)\ (1-\alpha_3)\ F_2F_3\ [F_{2\text{-}1} + (1-\alpha_3)\ F_{2\text{-}3}F_{3\text{-}1}F_1]} \Big/ \frac{}{[F_{1\text{-}2} + (1-\alpha_3)\ F_{1\text{-}3}F_{3\text{-}2}F_1]} \quad \ldots\ldots\ldots (1d)$$

and the net radiant heat exchange is

$$Q = S_{1\text{-}2}\ 0.174\ A_1 \left[\left(\frac{T_1}{100}\right)^4 - \left(\frac{T_2}{100}\right)^4\right] \quad \ldots\ldots\ldots (2)$$

Since Equation 1 involves the product of shape factors, an appreciable error will result for large area unless the product is integrated over both areas.

Application of Equation 2 to give the radiant transfer from an incandescent filament to a surface n is:

$$E_n'' = E'\ a_n\ F_2F_3F_4\ [F_{L\text{-}n} + (1-\alpha_x)\ F_{L\text{-}x}F_{x\text{-}n}F_1] \quad \ldots\ldots\ldots (3)$$

where subscript L refers to the filament and x to the combination of all surfaces excluding n.

$$F_1 = \frac{1}{1 - (1-\alpha_x)\ F_{x\text{-}x}} \quad \ldots\ldots\ldots (3a)$$

$$F_2 = F_4 = 1 \quad \ldots\ldots\ldots (3b)$$

$$F_3 = \frac{1}{1 - (1-\alpha_n)\ (1-\alpha_x)\ F_{x\text{-}n}F_{n\text{-}x}F_1} \quad \ldots\ldots\ldots (3c)$$

$$E' = mE \quad \ldots\ldots\ldots (3d)$$

where m is less than, or in the limit equal to unity, and E is the strength of the source. Absorptivity α must be used in Equation 3 rather than emissivity. F_2 and F_4 approach unity because of the very small area of the filament relative to surface n.

Only limited data are available for the absorptivity of wall surfaces for high temperature radiation, therefore the accuracy of Equation 3 is dependent on accurate data for the absorptivity. A value of 0.5 may be taken as a representative value for most light colored wall surfaces. Leopold** gives more detailed information.

The illuminating engineer in no case would select surface materials which absorb a high percentage of the visible portion of the spectrum. However, the air conditioning engineer wants the absorptivity of the panel surface to be high for the most efficient operating conditions. A fortunate characteristic of most light colored nonmetallic surfaces is a greater absorptivity for infrared radiation than for luminous. This feature may be enhanced by application of certain heat absorbing paints. The most satisfactory selective combination surface now avail-

**The Mechanisms of Heat Transfer, by C. S. Leopold. (*Refrigerating Engineering*, Vol. 54, July 1947, pp. 33-41.)

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able reflects about 75 percent of the visible component and 20 percent of the long wave length infrared radiation.

The use of reflectors to increase the shape factor of the cooling surface with respect to the source will increase, theoretically, the transfer rate of the panel. However, the increase may be small or nonexistent in the actual case because of resistance to conduction between the panel surface and the cooling medium. As the shape factor of the receiving surface with respect to the source increases, a greater quantity of energy will be concentrated on the surface. This increase of energy reception will tend to increase the temperature of the surface which will in turn, because of the greater temperature difference, increase the rate of heat transfer to the coolant. Unless the flow rate of coolant is increased, its temperature will rise, thereby reducing the ability to accommodate the increased energy received by the panel. Therefore, an increase in the shape factor will have only a limited increase on the capacity of the panel, unless an adequate increase in coolant flow rate is made.

Heat Balance. At equilibrium for a sealed enclosure, the energy received at an inside surface 1 by radiation and convection must exactly equal that transmitted through the wall by conduction. Hence

$$A_1C_1(t_1 - t_o) = \sum A_nF_{n-1}h_r'(t_n - t_1) + A_1h_{c1}(t_1 - t_a) + E_1'' \quad \ldots\ldots (4)$$

where C is the conductance of the wall between the inside surface and outside air.

Equation 4 does not include multiple reflections among the room walls, but E'' includes multiple reflections for the high temperature radiation.

A solution of Equation 4 gives

$$t_1 = \frac{(F_{1-2}h_{r(1-2)}t_2 + F_{1-3}h_{r(1-3)}t_3 + \ldots\ldots) + h_{c1}t_a + C_1t_o + E_1''}{(F_{1-2}h_{r(1-2)} + F_{1-3}h_{r(1-3)} + \ldots\ldots) + h_{c1} + C_1} \quad \ldots\ldots (5)$$

$$t_2 = \frac{(F_{2-1}h_{r(2-1)}t_1 + F_{2-3}h_{r(2-3)}t_3 + \ldots\ldots) + h_{c2}t_a + C_2t_o + E_2''}{(F_{2-1}h_{r(2-1)} + F_{2-3}h_{r(2-3)} + \ldots\ldots) + h_{c2} + C_2} \quad \ldots\ldots (6)$$

$$t_n = \frac{(F_{n-1}h_{r(n-1)}t_1 + F_{n-2}h_{r(n-2)}t_2 + \ldots\ldots) + h_{cn}t_a + C_nt_o + E_n''}{(F_{n-1}h_{r(n-1)} + F_{n-2}h_{r(n-2)} + \ldots\ldots) + h_{cn} + C_n} \quad \ldots (7)$$

where n is the number of surfaces used. The additional equation required for a simultaneous solution may be written by an energy balance on the ventilating air. Hence

$$t_a = \frac{h_{c1}t_1A_1 + h_{c2}t_2A_2 + \ldots\ldots + 0.24Wt_o + (E\text{-}E')}{h_{c1}A_1 + h_{c2}A_2 + \ldots\ldots + 0.24W} \quad \ldots\ldots\ldots\ldots (8)$$

Experimental Results

In order to verify the theoretical equations previously stated, it is necessary to obtain data and information so that a heat balance may be written on the enclosure and the panel. These data include: (1) temperature of room air; (2) temperature of air behind each enclosure wall; (3) inside surface temperature of each wall; (4) average temperature of panel; (5) panel coolant flow rate; and (6) the change in coolant temperature between panel entrance and exit. In addition, thermal conductance of each wall, combined radiation and convection coefficients for exterior surfaces, convection coefficients for internal surfaces, complete shape factor analysis, surface absorptivity characteristics, and the dissipation rate of the source must be known.

The thermal conductance of each of the enclosure walls was evaluated by application of thermal conductivity data for the particular construction. The combined film coefficients for the outside surfaces were approximated by data given in the ASHVE GUIDE, while inside film coefficients for convection from the surface to air at room temperature are approximately 1.1, 0.7 and 0.4 for horizontal plates facing up, vertical plates, and horizontal plates facing down respectively. The absorptivity of the white matt surface of the enclosure was taken as 0.5 and 0.93 for high and low temperature radiation respectively. The remaining requisite items were determined by experiment.

Experimental data pertinent to this subject were taken by the author at Purdue University.[□] In order to interpret the data relative to this subject, it is believed necessary to analyze the conditions under which the data were taken.

Experimental Room. The enclosure consists of a sealed room with the overall dimensions being 13.27 by 9.35 by 7.35 ft high with a 3.62 by 2.00 ft outset. The details are shown by Figs. 1 and 2. Fig. 1 also shows the location of the panel, area 105 sq ft. Fig. 2 depicts the outset as viewed from the center of the room. The source considered consisted of an incandescent bulb located approximately one foot below the ceiling at the center of the room.

The interior of the room is finished with plaster characterized by a white painted matt surface. The panel consists of water coils embedded in the ceiling and covering a major portion of the area. The coil

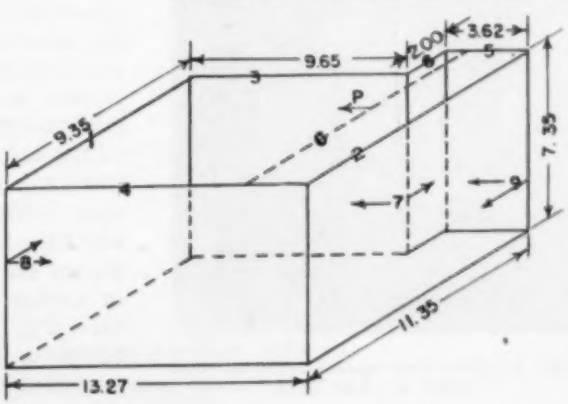

Fig. 1—Experimental room

□Master's Thesis, Merl Baker, February 1948, Mechanical Engineering Department, Purdue University, Lafayette, Ind.

Fig. 2—The room outset viewed from the center of the room

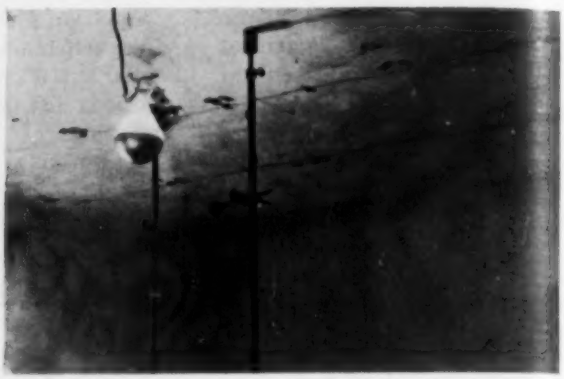

Fig. 3—Distribution of panel surface thermocouples

is constructed as a parallel series of one-half inch copper tubes, all connected into parallel headers. Approximately one-half the tube diameter is buried in the plaster.

Temperature Measurement. Twenty-five thermocouples were installed on the panel surface. These couples were copper-constantan located at the centers of rectangles approximately 22 by 26 in., thereby forming an approximate uniform pattern (Figs. 3 and 4). A uniform pattern is necessary if the average panel temperature is to be obtained by the arithmetic average of the point values. To be statistically exact, the rectangles must be replaced by squares, but in this case, the geometry of the surface dictated the use of rectangular sections in preference to square ones. Since the rectangles used were nearly square, the error may be considered negligible.

Each couple was placed into a prepared groove, the depth being approximately equal to one-half the diameter of the thermocouple junction, and was held in place by translucent tape firmly pressed over the junction. The thermocouple leads were held against the surface for a distance of about six inches. Six thermocouples were installed to measure the room air temperature near the ceiling, floor and mid plane of room. Initially many positions were checked, but it was found that the air for a particular horizontal plane remained essentially constant throughout the room except for positions very near the source, therefore, it was concluded that two vertical positions would be adequate. One vertical stand, located between the source and one interior wall remained fixed while the other one was frequently changed. Negligible fluctuations were observed among corresponding temperatures. The temperature gradient between the ceiling and floor was found to be less severe when a radiant source was employed than with an equally rated convective source. Additional thermocouples were employed to measure the temperature of the air behind each wall. An ice bath was used for the reference junction for all couples.

The calorimetric measurements for the panel coolant appear to be the most critical. The inlet and outlet coolant temperatures were continuously recorded by a thermocouple potentiometer combination with the flow rate being determined by a rotameter. The flow rate was kept low in order to obtain as great a temperature rise as possible, thereby contributing to the accuracy of thermal measurements. However, low rates were more susceptible to pressure fluctuations and created a greater non-uniformity of temperature across the panel surface.

Fig. 4—Two foot square reflector and 1000 w clear bulb

Source Location. The source was located at the center of the room, off center with respect to the panel, with the center of the filament being one foot below the ceiling for a 1000 watt bulb and somewhat less for smaller bulbs. When used, reflectors were supported in a horizontal plane approximately four inches below the filament center.

Four aluminum foil reflectors were employed: (1) one foot square plane; (2) two foot square plane; (3) one foot square plane with a two inch band; and (4) one foot square plane with a four inch band.

Convective Equivalent. The convective equivalent is defined as the hypothetical capacity of a convective source possessing a heat dissipation rate to the air and on to enclosure surfaces just sufficient to maintain the room air temperature at the same level as that resulting from a radiant source. The difference between the source strength and the convective equivalent is the energy portion which is ineffective in raising the room air temperature. Fig. 5 depicts the contrast between a convective and radiant source of the same strength.

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Table 1—Summary of Experimental Data

Test	Source: Rating Watts	Source: Glass Type	Source: Reflector	Shape Factor	Panel Temperature	Air Temperature: Midway	Air Temperature: Outside	Water Temperature Average	Heat Removed[a] by Panel Coolant
1	1050	Clear	None	0.406	72.4	79	68	63	1860
2	1050	Clear	1	0.615	76.6	82	72	67	1900
3	1050	Clear	2	0.726	78.7	82	68	68	1760
4	1050	Clear	3	0.718	77.6	82	73	67	2190
5	1050	Clear	4	0.827	78.2	81	75	67	1975
1	1050	Clear	None	0.406	72.4	79	68	63	1860
6	1000	Inside Frosted	None	0.406	71.5	80	74	63	1950
7	1000	Silver Base	None	0.850[b]	78.0	81	74	69	1910
1	1050	Clear	None	0.406	72.4	79	68	63	1860
8	500	Clear	None	0.406	68.5	73	61	63	1165
3	1050	Clear	2	0.726	78.7	82	68	68	1760
9	1050	Clear	2	0.726	87.7	86	65	84	—310
10	1050	Clear	2	0.726	99	91	74	100	—750
11	1050	Clear	2	0.726	111.8	95	70	119	—1740
12	1050	Clear	2	0.726	117.8	99	72	136	—2280

[a]Includes flow to back of panel and gain from storage.
[b]Assumed
1 One foot square plane
2 Two foot square plane
3 One foot square plane 2 inch band
4 One foot square plane 4 inch band

Table 2—Convective Equivalents

Test	Convective Equivalent Btu per hour	Percent of Source Input Ineffective in Raising Air Temperature
1	1262	65
2	1275	65
3	1353	63
4	1193	67
5	1248	65
1	1262	65
6	1136	67
7	1140	67
1	1262	65
8	696	59
3	1353	63
9	1849	55[a]
10	1940	62[a]
11	2570	65[a]
12	2645	73[a]

[a]Considering all input of panel goes into heating air.

Tables 1, 2 and 3 summarize the experimental data.

Confirmation of Equations. Equations 3, 4, 7, 8 may be checked by comparing resulting calculations with experimental data. Test one is calculated by the theoretical equations. Equation 3 is applied to compute the heat transfer to each surface resulting from that initially emitted by the source. Equations 7 and 8 are applied to obtain the surface temperatures of the enclosure.

Table 3—Wall Conductances Including Back (Outside) Side Film Coefficient

Surface		Conductance
7	(Combination 3 and 6, Fig 1)	1.41
8	(Combination 1 and 4, Fig 1)	0.114
9	(Combination 2 and 5, Fig 1)	0.675
w	(Window)	0.620
f	(Floor)	0.227
z	(Uncooled ceiling)	0.118

Table 4—Calculated Temperatures

$t_7 = 77.54$	$t_f = 76.60$
$t_8 = 78.58$	$t_c = 79.70$
$t_9 = 78.07$	$t_p = 72.40$
$t_w = 74.81$	$t_z = 79.00$

Table 4 gives a list of the calculated surface temperatures.

Heat Balance on Panel. Applying Equation 3, the energy received by the panel surface, directly or by reflection *from the incandescent filament,* is computed as follows:

$F_1 = 1.67$
$F_2 = 1.08$
$E_p'' = 0.7 \times 3590 \times 0.5 \times 1.08 [0.406 + (1 - 0.5) 0.594 \times 0.192 \times 1.67]$
$E_p'' = 678$ Btuh (From incandescent filament)

The energy emitted from bulb glass as low temperature direct radiation may be approximated as follows: A value of 286 Btuh is obtained by assuming that approximately 20 percent of the filament energy is dissipated at or within the globe and by assuming that 40 percent of this quantity is dissipated by radiation. Since the shape factor with respect to the bulb is 0.406, the energy received by the panel from the bulb glass is:

$0.406 \times 286 = 115$ Btuh (from bulb glass)

The total direct and indirect radiation received by the panel from filament and glass bulb is:

$Q_L = 678 + 115 = 793$ Btuh

The energy received by the panel by convection is:

$q_c = h_c A\ (t_{sr} - t_p)$
$q_c = 0.55^{\dagger\dagger} \times 105\ (81.0 - 72.4) = 495$ Btuh

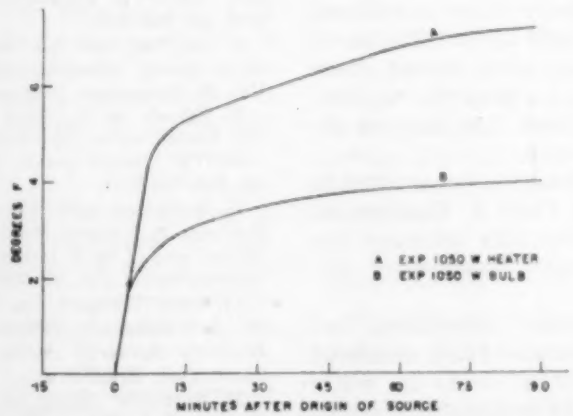

Fig. 5—Temperature rise of room air

††Evaluated in accordance with Heating, Ventilating, Air Conditioning Guide 1948.

The energy received by the panel by radiation from enclosure is:

$$q_r = A_p h_r \ (MRT - t_p)$$
$$q_r = A_p h_r \ (F_{p-1} \Delta t_1 + F_{p-2} \Delta t_2 + \ldots\ldots F_{p-n} \Delta t_n)$$
$$q_r = 105 \times 1.05 \ [0.187 \ (77.54 - 72.40) + 0.308 \ (78.58 - 72.40) + 0.137 \ (78.07 - 72.40) + 0.066 \ (74.81 - 72.40) + 0.307 \ (76.60 - 72.40)]$$
$$q_r = 558 \text{ Btuh}$$

The heat lost to coolant from the room:

$$q = Q - q_b$$
$$q_b = AC \ (t_w - t_b)$$
$$q_b = 105 \times 0.119 \ (76 - 63) = 163 \text{ Btuh}$$

where

Q = the heat gained by the coolant.
q_b = the gain from the back of the panel.
$q = 1860 - 163 = 1697$ Btuh

Total heat received by panel	793	
	495	
	558	
	1846	Btuh
Total heat lost by panel (By experiment)	1697	
Unbalance	149	Btuh

Room heat balance		
Conduction loss (Calculated)	1595	
Panel loss	1697	
Radiation loss (Glass window)	145	
	3437	Btuh
Input (Electrical input to filament)	3590	Btuh
Unbalance	153	Btuh

The energy balances are remarkably close considering the assumptions made relative to the condition of steady state transfer and in the evaluation of some physical properties.

Conclusions

Applications of the theoretical equations require the knowledge of the air temperature back of the various walls, the temperature of the air in the enclosure and the conductances of the various walls. These conductances may be obtained by use of tables in the HEATING, VENTILATING, AIR CONDITIONING GUIDE, and air temperatures determined from design data.

Application of Equations 3 and 4 reveals that a room with one-half the surface area exposed to the outside generally possesses enclosure surfaces at temperatures low enough to absorb and remove about 50 percent of the total lighting load. This portion of the lighting load is removed from the structure without heating the air; therefore, a consideration that all the electrical lighting load is transmitted to the air may result in an over design of the cooling equipment.

The theoretical equations may also be employed to confirm the results of Table 2, that by the utilization of ceiling cooling panel, the percent of the lighting load directly removed from the conditioned space without heating the air is increased to approximately 65 percent. The calculated heat balance for both the room and panel for test 1 are closely duplicated by the measured quantity.

Experimental data recorded in Table 2 reveal that the percent of filament input (of a 1000 watt clear bulb utilizing a 2 ft plane reflector) which is ineffective in heating the room air varies from 63 at a panel temperature of 78.7 F to 73 at a panel temperature of 117.8 F. For the same source and reflector for panel temperatures of 87.7 F, 99 F, and 111.8 F, the percentages are 55, 62, 65 respectively. Because of the possibility of not having steady state transfer, it is believed that the extreme values are not justified and an average value of 65 percent is representative regardless of the panel temperature.

Table 2 also shows that for this particular test the percent of the filament input that is ineffective in heating the air is independent of the type of reflectors used, although the shape factor range considered varied from 0.406 to 0.827. The panel surface temperature varied from 72.4 to 78.2 F. However, as previously explained, this may not always be the case.

The type of bulb is not significant as shown by Table 2. However, as the size of the bulb decreases the rate of removal apparently decreases.

The favorable comparison between the calculated and measured panel absorption tends to verify the hypothesis that multiple reflection must be considered when dealing with high temperature energy sources.

Acknowledgment

Equipment used in obtaining the experimental data for this project was purchased with funds granted by the Copper and Brass Research Association to the Purdue University Engineering Experiment Station.

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Vaneaxial Fan Fundamentals†

By Raymond Mancha*, Pittsburgh, Pa.

SUMMARY—The author describes the characteristics of vaneaxial fans and the factors affecting fan design and discusses the limiting values of these factors. Explanations are given of the action of rotor, vanes, fan blades and stack upon the air in contact with them. Equations and design curves are included to assist the designer in obtaining a quick analysis of design possibilities for a given fan duty.

General Description

THE principal parts of the conventional vaneaxial fan equipped with stationary vanes consist of the *inlet section*, *stage* and *stack* as illustrated by Fig. 1.

Inlet section is equipped with a cowling and cowling supports and is designed so as to permit air to enter the stage with the airflow pattern upon which the design depends.

Fan *stage* consists of the rotor and stationary vanes; it is within the stage that the air is compressed and delivered to the stack with the proper airflow pattern to permit efficient stack action.

Stack is equipped with a discharge cowling and cowling supports and is designed to efficiently convert the air speed from the fan outlet to the desired value. Thus, a blowing fan can supply air to the system at the desired air speed and an exhaust fan can discharge air to the atmosphere with low kinetic or waste energy.

Capacities and Speeds

For a particular size vaneaxial fan equipped with stationary vanes, customary design practice dictates a design air volume and a design fan speed for any particular fan pressure. When passing the design air volume, conventional design provides that the air enters and leaves the fan stage with equal stage pressure increase and with pure axial air flow of uniform and equal speed at all radii. Design speed is the fan speed corresponding to design air volume at design fan pressure.

For geometrically similar fans, design air volume varies directly as the square root of design fan pressure and directly as the square of the fan diameter. Likewise, design fan speed varies directly as the square root of design fan pressure but inversely as fan diameter. For classifying a family of geometrically similar fans as to capacity and speed, it is therefore convenient to refer to a one foot diameter prototype handling air weighing 0.075 lb per cu ft and developing a fan pressure of one inch water column. Thus, design air volume is called design unit volume, and design fan speed is design unit speed.

General classification of vaneaxial fans as to capacity or speed is entirely arbitrary. However, it might be useful to refer to design unit volumes of 1000 cfm or under as low capacity, from 1000 cfm to and including 2000 cfm as intermediate capacity and above 2000 cfm as high capacity. Similarly, one might classify design unit speeds of 3000 rpm and under as low speed, from 3000 rpm to and including 4000 rpm as intermediate speed and above 4000 rpm as high speed.

The ratio of the fan rotor diameter at the blade root to the rotor diameter at the blade tip is referred to as hub ratio. This ratio is preserved throughout the size range of any family of geometrically similar fans.

Rotor Action

The total work done upon the air passing through a vaneaxial fan rotor is partially realized by a mechanical energy addition to the air equal to the algebraic sum of the static pressure energy increase and the rotative energy decrease or increase, depending upon the location of the stationary vanes.

With guide vanes ahead of the rotor, the rotor both compresses the air and converts rotative energy to pressure energy. With straightener vanes behind the rotor, the rotor both compresses the air and imparts rotative energy to the air.

Stationary Vanes Action

Guide vanes, located immediately ahead of the rotor, impart an initial rotation to the air entering the rotor. This initial air rotation is counter in direction to that of the rotor and ½ of an amount just sufficient to permit the air to leave the rotor axially.

Straightener vanes, located immediately behind the rotor, partially recover the rotative energy imparted to the air by the rotor and reestablish axial flow to the air leaving the vanes.

Any tendency toward radial air flow due to the rotation imparted to the air by the guide vanes or the fan rotor is negligible for conventional design. This tendency is ignored because the radial pressure gradient of the rotating air mass very nearly provides the centripetal force distribution necessary for radial equilibrium.

Multi-Staging

In the multi-staging of vaneaxial fans, it is customary to disregard small air density changes due to compression and to apply identical stages in series. Sometimes, however, the pressure range or compression dictates successive stages of different design. This is frequently the case in design for high fan pressures and always the case for vaneaxial or *turbo* compressor design.

It is theoretically possible to eliminate the use of vanes with a two-stage fan consisting of two dif-

†*Abstract:* The development of formulas and the details of design are described more fully in Appendices I and II to be published with the paper in the 1949 TRANSACTIONS.

*Vice president, Ventilation, Joy Mfg. Co.

ferently designed rotors operating in tandem with opposite rotation. The failure of this design to attain popularity is probably due to mechanical complications and commercial aspects.

Stack Action

Fan pressure refers to the amount by which the absolute reference pressure (static or total) at the fan outlet exceeds the absolute total pressure at the fan inlet. *Stage pressure* is the amount by which the absolute total pressure at the stage outlet exceeds the absolute total pressure at the stage inlet. Losses between the fan inlet and the stage inlet can be neglected because of accelerated air flow. Therefore, the amount by which stage pressure exceeds fan pressure is referred to as stack loss. The ratio of stack loss to the velocity pressure referred to stage axial air speed is called stack coefficient and is herein designated by the symbol K.

Stack coefficients will vary between the limits of zero and unity, depending upon the efficiency of pressure conversion within the stack, which in turn depends upon the stack design. Crediting a fan with the stage total pressure corresponds to assigning a stack coefficient K equal to zero since losses beyond the stage are excluded. Crediting a fan with the stage static pressure corresponds to a stack coefficient K equal to unity since the fan is thus charged with losses beyond the stage equal to one stage velocity head. There is no need for a stack coefficient to exceed unity because such a stack would be worse than no stack at all.

Minimum Speed Fans

For a particular design capacity, hub size and stack, fan unit speed can be selected within limits by varying blade widths and angles. The upper speed limit is dictated by efficiency, noise and mechanical considerations, whereas the lower speed limit is reached when further increase of blade width and angle at the blade root or hub fails to provide the required rotor pressure.

With constant axial air speed at all radii it becomes apparent that the blade speed relative to the air must be lowest at the hub where the blade tangential speed is lowest. To compensate for the low blade to air speed at the hub, the blade width must be increased when the available range of blade angle variation alone is inadequate to compensate for the relative speed reduction.

The action of the air passing through the fan rotor is analogous to air passing through a diffuser in that the air speed relative to the rotor blades or diffuser is reduced with an accompanying increase in static pressure in accordance with Bernoulli's theorem. The same is true of the air traversing straightener vanes whereas with guide vanes the air is accelerated at the expense of and with an accompanying reduction of static pressure.

The static pressure increase from the leading to the trailing edge of an airfoil in retarded flow is limited to a fraction of the velocity pressure of the approaching air relative to the airfoil. This fraction or *loading* is approximately 0.63, according to the results of experimentation by Curt Keller[1] with a single symmetrical airfoil in retarded flow at zero degrees attack angle. Above this critical loading air flow, separation occurs at the airfoil surface near the trailing edge and the airfoil *stalls*.

Since vaneaxial fan rotor blades and straightener vanes are actually airfoils in retarded flow, rotor and straightener vane hub loading are influential factors determining minimum design unit speed. Mutual blade and vane effects are more reflected in restriction of maximum airfoil lift than in reduction of lift for any particular attack angle.

Drive Location

The performance of vaneaxial fans is impaired to varying degrees by obstructions in the air stream located either just ahead of the fan stage or in the fan stack just beyond the stage. The sensitivity to such obstruction increases with the fan design or operating capacity.

[1] *Axial Flow Fans*, by Keller-Marks.

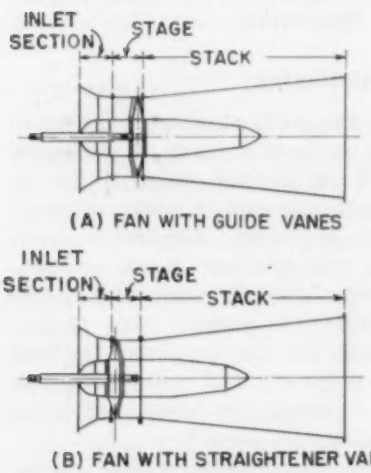

Fig. 1—Principal parts of vaneaxial fans

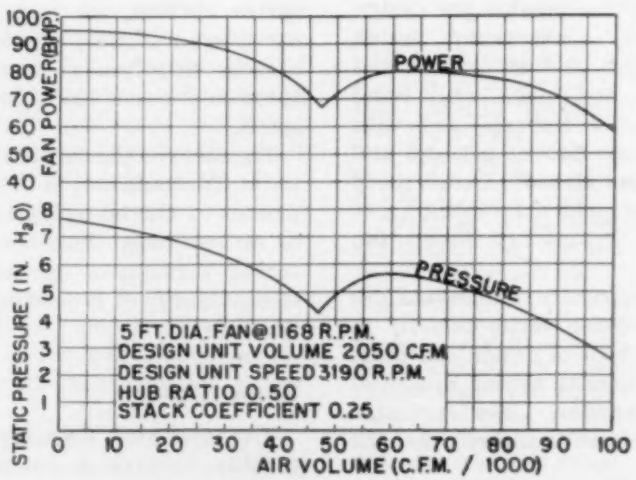

Fig. 2—Typical pressure and power curves for vaneaxial fan *For high capacity, intermediate speed fan equipped with straightener vanes*

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High capacity fans are most affected and low capacity least affected.

Obstructions ahead of the fan stage that merely distort the flow pattern of the air entering the stage tend to reduce the stage pressure and may increase the fan power, thereby reducing fan efficiency.

Obstructions ahead of the stage (such as exposed belt drives) that impart rotation to the air in the direction of fan motion reduce the stage pressure and power quite noticeably for small amounts of pre-rotation. But by approximately the same percentage these obstructions result in little or no effect upon fan efficiency. Fans with straightener vanes are more affected by pre-rotation of the air entering the stage than are fans with guide vanes.

Obstructions of any sort located in the fan stack just beyond the stage have the effect of reducing the stack pressure conversion efficiency without affecting the fan power, thereby reducing the fan efficiency by the same percentage that the fan pressure is reduced.

When vaneaxial fans are driven with belt drives between the fan rotor and the power unit, best results are obtained with drives located ahead of the fan inlet with the belt or ropes covered in the air stream by two streamlined fairings. Thus pre-rotation of the air entering the fan stage is avoided with the minimum possible obstruction to air flow. The fairings are thereby located in low speed air, and the air after passing over the fairings is accelerated upon entering the fan inlet. This acceleration tends to iron out any flow pattern distortion resulting from the obstruction offered by the fairings.

Belt fairings located in the fan stack are in high speed air undergoing deceleration which tends to exaggerate the flow pattern distortion caused by the fairings.

Sometimes it is possible to locate a vaneaxial fan at an angle to the duct approaching or leaving the fan and to extend the drive shaft outside the air stream. This permits the use of short, uncovered belt drives or direct connection to the power unit when speeds allow. Direct connection to the power unit eliminates drive losses. Air duct bends ahead of the fan should be equipped with properly designed turning vanes to direct air into the fan inlet with uniform distribution.

Frequently it is practicable to locate the power unit inside the fan cowling and mount the fan rotor on the shaft of the power unit. Or else separate fan bearings can be provided and the rotor driven by means of suitable coupling to an externally located power unit.

Stall Characteristics

Fig. 2 illustrates typical pressure and power curves for a high capacity, intermediate speed vaneaxial fan equipped with straightener vanes. These curves are for a 60-in. diameter fan at 1168 rpm having a design unit volume of 2050 cfm, design unit speed of 3190 rpm, hub ratio 0.50 and stack coefficient 0.25.

The design volume for this fan at the operating speed is 93,800 cfm which is calculated from the design unit speed and design unit volume obtained by formulas 1 and 2.

The cusps which occur in the two curves at a volume of approximately 47,000 cfm occur for reasons analogous to the reasons that an airplane stalls when attempting to climb too rapidly. This point of operation is therefore referred to as the stall point because at this point the fan blades actually go into an aerodynamic stall like the airplane wing.

The blade of the vaneaxial fan, as well as the airplane wing, acts as an airfoil when it is moved relative to the air and inclined to the line of relative motion at the necessary angle of attack. This angle effects the required force or lift exerted upon the blade or wing in a direction normal to the line of relative motion.

Fig. 3 shows how lift varies with angle of attack and is qualitatively typical of fan blade airfoil shapes. It is apparent that there is a value of angle of attack beyond which the lift ceases to increase and falls off precipitously. This maximum angle of attack marks the stall point for the airfoil in question and must not be exceeded for efficient lift.

The angle that the fan blade makes with the piane of rotation is greatly influenced by the ratio of the axial air speed to the tangential blade speed at any particular radius

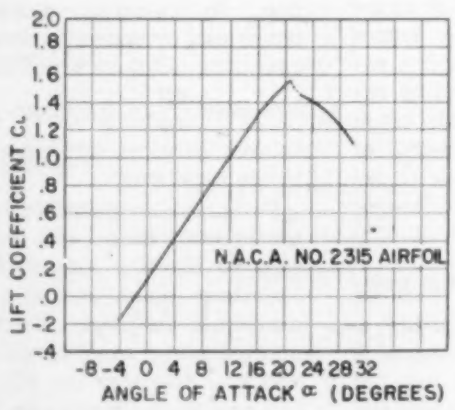

Fig. 3—Variation of lift with angle of attack

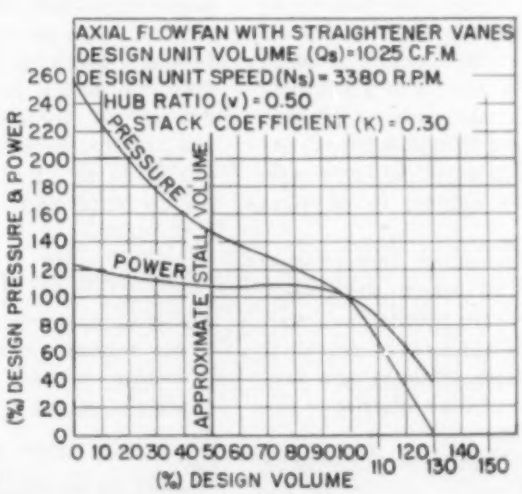

Fig. 4—*(Right)* Pressure and power curves and stall volume for vaneaxial fan

along the blade. This ratio, called discharge coefficient, is minimum at the blade tip where the blade tangential speed is highest and maximum at the blade root or hub where tangential blade speed is lowest for conventional design embodying constant axial air speed at all radii.

The design angle of attack along the fan blade is comparatively constant. It is a small percentage of the total angle between the blade and the plane of rotation; this total angle in turn is of the same order of magnitude as the angle whose tangent is expressed by the discharge coefficient. Therefore, changes in angle of attack are disproportionate, percentagewise, to corresponding changes in discharge coefficient. This disproportionality increases with increased design discharge coefficient.

Obviously a reduction in discharge coefficient increases angle of attack. Therefore, by reducing the air volume handled by a vaneaxial fan operating at constant speed, the discharge coefficient is reduced at all radii and the angle of attack is increased, especially so at the hub where the discharge coefficient is maximum.

Stalling usually starts at the hub and progresses radially outward along the blade to the tip; this condition is illustrated by the cusps in the pressure and power curves of Fig. 2.

Stalling is physically manifested by pressure instability, change of fan sound to a pulsating roar and severe whirling of the air immediately ahead of the fan rotor. At volumes below the stall point air volume, the pressure becomes steady but the roar and air whirling ahead of the rotor, continue. When operating in this manner, the fan efficiency is very low. However, the fan is not subject to physical harm.

Practically speaking, fans of high capacity or low speed design, or both, have a more pronounced stall characteristic than low capacity, high speed fans, because the former types are of high discharge coefficient design.

Fig. 4 illustrates this point with pressure and power curves for an intermediate capacity, intermediate speed family of vaneaxial fans having a design unit volume of 1025 cfm, a design unit speed of 3380 rpm, a hub ratio of 0.50 and a stack coefficient of 0.30. For this fan the stall characteristic is only faintly discernible, manifesting itself by a slight dip in the pressure and power curves.

Considering low first cost and quietness of operation, high-capacity low speed fans are desirable. Fortunately the volume range between the design point and stall point is sufficiently wide to permit application of high capacity, low speed fans to most ventilation systems without likelihood of the fan ever operating in a stall. Even in the case of ventilating systems of variable resistance, in installations such as mines, it is usually possible to so select the fan that stalling at a future date will be avoided.

Pressure-Volume Curve

In addition to the fan stall characteristics already discussed, the shape of the fan pressure-volume characteristic curve and influential factors affecting the shape should be considered.

Air volume variation *above* the design volume for a vaneaxial fan operating at constant speed and blade position is accompanied by *decrease* in the air angle of attack along the blade, causing *reduction* in stage pressure. Air volume variation *below* the design volume under the preceding conditions would be accompanied by *increase* in the air angle of attack and would cause *gain* in stage pressure. The degree to which the stage pressure can exceed the design point value by reducing air volume depends upon the margin between the value of the blade air attack angle α at the design point and the maximum or stall value.

The slope $\frac{dP}{dQ}$ of the stage pressure-volume curve (P = pressure, Q = air volume) at design volume is therefore related to the rate of change of attack angle with volume, $\frac{d\alpha}{dQ}$, at the same point. For equal percentage air volume variation, the resulting change in blade air attack angle α is greater for high values of discharge coefficient than for low values, as already explained under the subhead Stall Characteristics.

It follows, therefore, that for a particular design volume and pressure the slope of the stage pressure-volume curve $\frac{dP}{dQ}$ at the design point can be increased or decreased, respectively, (1) given fan diameter and speed—by increasing or decreasing hub diameter; (2) given fan diameter and hub size—by reducing or increasing fan speed; (3) given fan speed and hub diameter—by decreasing or increasing fan diameter.

The effect of pressure losses beyond the stage is to produce a fan pressure-volume curve that is lower in pressure and steeper in slope than the stage pressure-volume curve for all air delivery volumes from the volume of maximum stage pressure to free delivery. The discrepancy in both pressure and slope increases with air delivery volume.

Power-Volume Curve

A constant speed vaneaxial fan with blades at design pitch or setting has a power-volume curve that can be designed to reach a maximum at the point of design volume or at any other desired operating point within the acceptable operating volume range above the stall region.

Useful air power, PQ, is the product of stage pressure P and air volume Q at any point of fan operation. By calculus, maximum air power can be shown to occur at the operating point at which the negative slope $\frac{dP}{dQ}$ of the stage pressure-volume curve is numerically equal to the ratio P/Q.

For practical purposes fan brake horsepower (bhp) can be considered to reach a maximum at the operating point corresponding to maximum air power. Therefore by manipulating fan diameter, speed and hub diameter the fan designer can provide for maximum fan power input to occur at, below, or above the design volume. The fan power-volume curve will, however, be flat-

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ter than the useful air power-volume curve due to the deterioration of stage efficiency for operation above or below design volume.

Adjustable Fan Blades

It is possible to adjust the blade setting with vaneaxial fans, thereby altering the fan's characteristics to suit changing pressure-volume characteristics of the ventilation system. Within a limited range of angles the blades continue to operate satisfactorily with stationary unaltered guide or straightener vanes.

Fig. 5 illustrates the point in question by presenting power and pressure curves for a 36-in. diameter vaneaxial fan with straightener vanes having a design unit volume of 2050 cfm, a design unit speed of 3190 rpm, a hub ratio of 0.50 and a stack coefficient of 0.25. The normal or design blade position is the No. 4 position and the fan in question operates at 1760 rpm.

It should be noted that whereas the design volume of this fan is 30,500 cfm, stalling occurs at 13,000, 15,400 and 17,400 cfm for blade positions Nos. 1, 4 and 7 respectively. There is a difference of 6 degrees only, in blade angle between positions 1 and 4 and 4 and 7, which illustrates how sensitive the fan performance is to change in blade position.

The principal purpose of blade adjustment is to maintain high fan efficiency in the face of change in the resistance of the ventilation system. The adjustable feature is not intended for major change in air volume when the fan operates on a system of fixed, unvarying resistance.

Fig. 5 illustrates the previous statement that stall tendency becomes more pronounced with increased discharge coefficient since discharge coefficient is increased with an increase in blade angle.

Series and Parallel Fan Operation

Any two fans, similar or dissimilar, can be operated in series provided that each fan is operated at a speed that will permit passage of the air volume required by the ventilation system.

The parallel operation of two fans requires not only the air volume capacity required by the ventilating system, but each fan must operate at a point on the characteristic fan pressure-volume curve where the slope of the curve is negative. The possibility of one fan blowing air back through the other fan is thereby eliminated, resulting in dependable, steady, parallel operation. Vaneaxial fans are well suited to parallel operation.

The question arises at times as to which type fan operation will produce the highest air volume for a particular ventilation or duct system having a resistance of $\frac{P}{Q^2}$ units.

Fig. 6 represents two similar fans operating at the same speed. Curve A is the pressure-volume characteristic of each fan. The fans selected are merely of a type having the negative slope pressure-volume characteristic and can be either centrifugal or vaneaxial fans.

Curve B of Fig. 6 is the composite or resultant pressure curve for the two fans operating in series. Curve C is the composite curve for the two fans operating in parallel. On the pressure-volume curve D for the duct system of resistance $\frac{P}{Q_o^2}$ units, curves B and C intersect at O, the point at which the two fans will pass the same volume when operated both at the same speed whether operating in series or in parallel.

Duct systems of higher resistance, Curve E, will favor series fan operation. Duct systems of lower resistance, Curve F, will favor parallel fan operation.

This same type analysis is valid for series or parallel operation of dissimilar fans or for similar fans at different speeds or blade positions.

Fan Design Curves

The fan design curves, Figs. 7 and 8, are based upon a design procedure that strikes a practical balance between fan efficiency, speed and experimentally established permissible hub loadings. These fans are designed to a constant value of 0.80 for the product of blade lift coefficient and ratio of blade width to

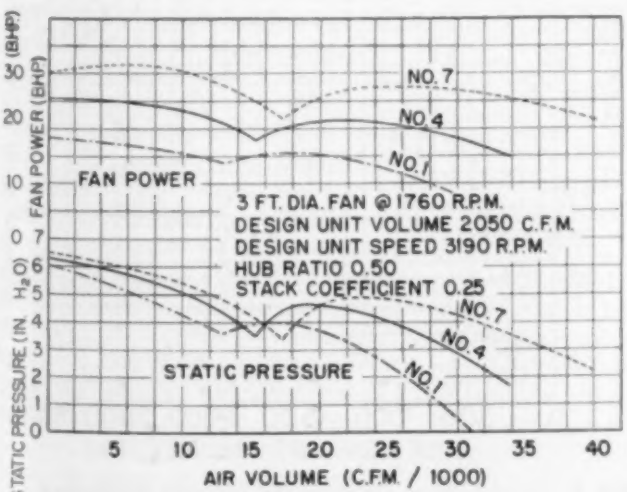

Fig. 5—*(Left)* Pressure and power curves for three fan blade positions

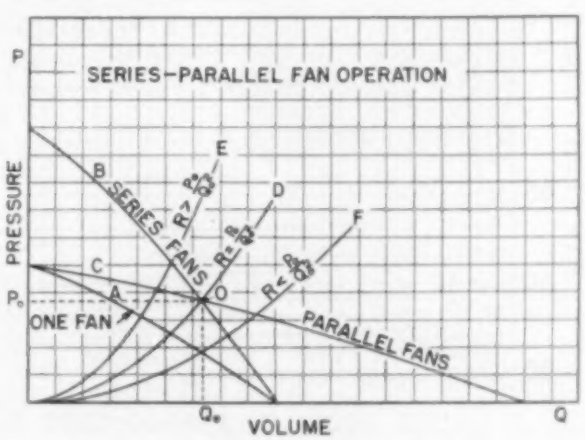

Fig. 6—Series-parallel fan operation

blade spacing at the blade root or rotor hub. This procedure observes a 0.60 loading limit for the fan rotor, throughout the entire range of fans shown. The 0.60 loading limit is also observed for the straightener vanes for all fans shown except those in the region of extremely low design capacities. For the latter, such liberties are permissible because of the relatively low rotational energies in the air entering the straightener vanes.

The aforementioned curves disregard the effects of hub windage and bearing losses. There is no modification for boundary layer effects which are negligible for all fans but those with extremely short blades.

An airfoil in an airstream is acted upon by a force called drag in the direction parallel to the relative air flow, and a force called lift in the direction normal to the relative air flow. The fan selection curves are based upon assumed airfoil drag to lift ratios of 0.03 and 0.06 for the rotor blades and stationary vanes, respectively.

Fan design curves, Figs. 7 and 8 for fans with either guide or straightener vanes, have fan efficiency and design unit speed plotted as ordinates against design unit volume as abscissas, for different values of stack coefficient K and different values of hub ratio.

Examination of the aforementioned curves shows fans with guide vanes to be inherently fans of lower speed than fans with straightener vanes. Since guide vanes expand the air and straightener vanes compress the air, the reverse might be expected owing to the greater compression required of the rotor with guide vanes. However, guide vanes impart to the air rotation which is counter to the direction of the rotor, thereby maintaining, between the air and the fan blade, the relative velocities necessary to provide the extra rotor pressure at reduced speed.

These curves further show that high fan efficiency and low blade tip speed, both desirable features, can be incorporated in the same design. Both are best effected with large, slow fans of low design unit volume and high hub ratio.

It is also apparent that for a particular design capacity, Q_s and stack coefficient K, there is a definite value of hub ratio v for maximum fan efficiency. Departure above or below this optimum ratio lowers fan efficiency and reduces or increases, respectively, fan speed.

Use of Fan Design Curves

To design a fan of diameter D ft to pass Q cfm of air weighing w lb per cu ft, at a fan pressure P in. water column and a fan speed of N rpm, the fan design unit volume Q_s and unit speed N_s are calculated by the following equations:

$$Q_s = \frac{3.65Q}{D^2\sqrt{\frac{P}{W}}} \quad \text{.....(1)}$$

$$N_s = \frac{3.65ND}{\sqrt{\frac{P}{W}}} \quad \text{.....(2)}$$

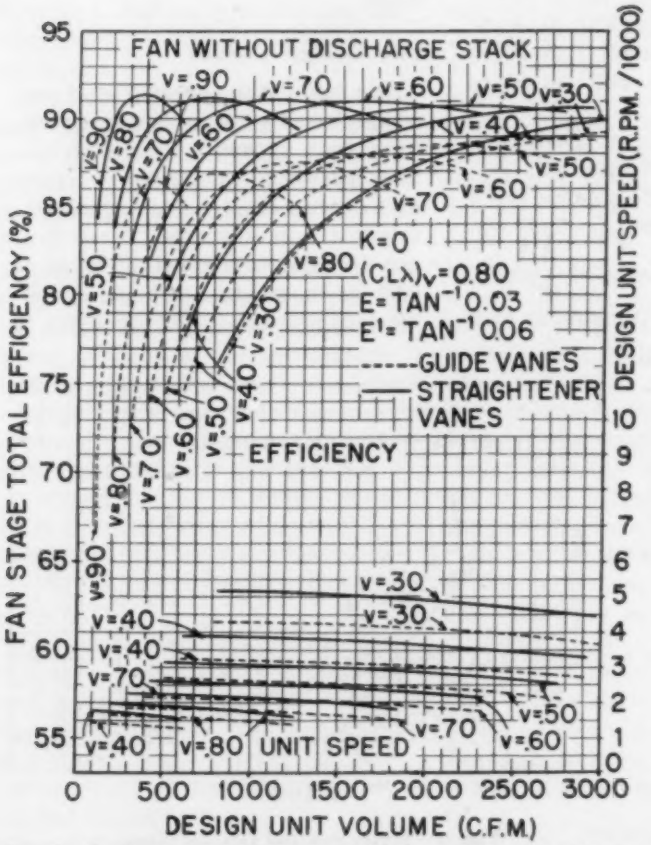

Fig. 7—Design curves for vaneaxial fan without discharge stack

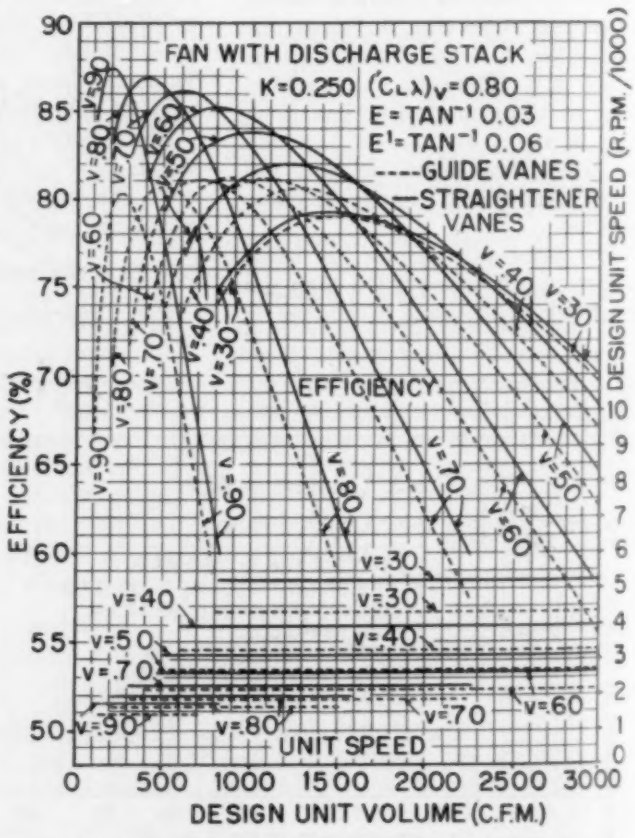

Fig. 8—Design curves for vaneaxial fan with discharge stack

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Having thus calculated design unit volume and design unit speed, reference can be made to the fan design curves, Figs. 7 and 8, for allowable hub ratios v and approximate fan efficiency for two values of stack coefficient K.

One example is the design of a 5-ft diameter fan to pass 100,000 cfm at 6 in. water gage with air weighing 0.065 lb per cu ft using a stack with a 0.25 coefficient, having fan efficiency as the sole objective. Using Equation 1, design unit volume is calculated at 1520 cfm. Reference to the curves of Fig. 8 for 0.25 stack coefficient shows the most efficient fan to be a fan with straightener vanes having a hub ratio of approximately 0.50 and a resulting fan efficiency of approximately 81.6 percent exclusive of bearing and hub friction losses. This fan would have a design unit speed of 3140 rpm and would thus operate at a fan speed of 1655 rpm.

The most efficient fan shown with guide vanes would be approximately 80 percent efficient with a 0.40 hub ratio, a design unit speed of 3300 rpm and a fan speed of 1740 rpm. The use of a 0.50 hub ratio with guide vanes would reduce the design unit speed to 2640 rpm and the fan speed to 1390 rpm but would also reduce the fan efficiency to 79 percent which is so slight as to be possibly justified by the marked speed reduction.

It is interesting to note that the use of a 0.60 hub ratio with straightener vanes would result in approximately the same fan efficiency and speed as the use of a 0.50 hub ratio and guide vanes.

It must be remembered that the fan design curves, Figs. 7 and 8, are suggested for convenient reference only, and represent but a small percentage of the permissible designs within the 0.60 hub loading limit.

CARRIER, HARDING, FARNHAM HONORED BY BUFFALO SOCIETY

W. H. Carrier, L. A. Harding and Roswell Farnham were among the 15 past presidents honored on September 15, 1949 by the *Engineering Society of Buffalo, Inc.* The occasion was the Seventh Annual Dinner honoring the past presidents and was held at the Hotel Markeen in Buffalo. Nearly 250 persons attended. Guest speaker was H. C. Lautensack, president of the Buffalo Chamber of Commerce.

Dr. Carrier was unable to attend but sent a telegram which was read to the group. Dr. Carrier served as the *E.S.B.'s* second president in 1913-14. The *E.S.B.* was formed in 1912, as a re-organization of an earlier society, the Engineering Society of Western New York, which was created in 1894. Dr. Carrier is a charter member of the earlier society. He served as president of ASHVE in 1931.

Messrs. Harding and Farnham were present, sitting next to each other. Mr. Harding is a past president of ASHVE (1930) and was president of *E.S.B.* in 1933-34. Mr. Farnham served on ASHVE Council from 1927-33 and was president of *E.S.B.* in 1930-31. Mr. Carrier is chairman of the board, Carrier Corp., Syracuse, N. Y.; Mr. Harding is a consulting engineer in Buffalo; and Mr. Farnham is district manager, Buffalo Forge Co.

The 15 past presidents were introduced by J. Lloyd Walker. The 1949-50 officers of *E.S.B.* were installed by G. T. Parker, Jr.

CHICAGO MAN ELECTED PRESIDENT OF *ASME*

James D. Cunningham, Chicago, Ill., newly elected president of the *American Society of Mechanical Engineers* for 1950, will take office at the end of the *ASME* 1949 Annual Meeting to be held in New York City, November 27 to December 2, 1949. He will succeed James M. Todd, New Orleans, La.

Other officers elected are: Frank M. Gunby, Boston, Mass.; John C. Reed, Lewisburg, Pa.; Albert C. Pasini, Detroit, Mich.; Samuel H. Graf, Corvallis, Ore.; Benjamin P. Graves, Providence, R. I.; and Thomas E. Purcell, Pittsburgh, Pa.

PUBLICATIONS LIST AVAILABLE

A convenient list of publications dealing with homes, home equipment and construction is now available at the National Bureau of Standards, U.S. Department of Commerce, Washington 25, D. C. The list is Letter Circular 951, List of Published Material Relating to Home Building and Maintenance, and is intended to be of assistance to prospective home owners, home builders and others interested in building materials, equipment and practices used in dwelling construction.

VOORHEES NAMED TECHNICAL SECRETARY OF *NWAHACA*

Guy A. Voorhees, application engineering director of the *National Warm Air Heating and Air Conditioning Association*, Cleveland, Ohio, has recently been assigned the additional duty of Technical Secretary. Mr. Voorhees recently received an honorary degree in Mechanical Engineering from Michigan State College.

Guy A. Voorhees

Besides being a member of the ASHVE, he is also a member of *ASME* and the Indiana Society of Professional Engineers.

Standardization of Thermodynamic Properties of Moist Air†

Final Report of Working Subcommittee, International Joint Committee on Psychrometric Data

By John A. Goff*, *Chairman*

Introduction

THE inaugural meeting of the International Joint Committee on Psychrometric Data (IJCPD) was held in Boston, Massachusetts, on January 23, 1945, in connection with the 51st Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS. In November, 1945, the late J. Herbert Walker, temporary chairman, appointed a Working Subcommittee, of which the writer was privileged to be Chairman, to assist in the drafting of an agenda for the next meeting of his Committee.

The Working Subcommittee met first in Philadelphia on February 16, 1946. The minutes of this first meeting were used as an agenda for the second meeting held in Washington, D. C., on November 26, 1946, and attended by a number of interested non-members including representatives of the British Meteorological Office, the British Commonwealth Scientific Office, and the National Bureau of Standards. The minutes of this second meeting contain an item instructing the chairman to prepare a report of the Working Subcommittee's deliberations.

A preliminary draft of the above-mentioned report was submitted, under date of April 14, 1947, to Mr. Walker who offered several valuable criticisms and suggestions for incorporation in the final draft. Not until now, however, has the writer found it possible to prepare this final draft. Meantime the preliminary draft has served a very useful purpose as will be noted.

At the first full session of the Aerological Commission of the International Meteorological Organization (IMO) which met in Toronto, Canada, during August, 1947, there was appointed a Subcommission I on Functions and Tables under the chairmanship of Prof. P. A. Sheppard, Department of Meteorology, Imperial College of Science and Technology, University of London. By its terms of reference this Subcommission was charged with the task of recommending a set of self-consistent, internationally acceptable values of physical functions and constants of common use in meteorological practice. It is exceedingly gratifying to note the extent to which those recommendations of the Subcommission having to do with the thermodynamic properties of moist air agree with the recommendations of the Working Subcommittee as set[1] forth in the preliminary draft referred to in the preceding paragraph. The former were adopted officially by the Twelfth Conference of Directors of the International Meteorological Organization sitting in Washington, D. C., September, 1947.

The report of Professor Sheppard's Subcommission is soon to be printed and widely distributed by the International Meteorological Organization as Document No. 71 of its Aerological Commission. It seems especially appropriate, therefore, to publish the final report of the Working Subcommittee of the International Joint Committee on Psychrometric Data more or less simultaneously. There is ample justification for the *ASME* Special Research Committee on Properties of Gases and Gas Mixtures to sponsor the presentation and publication of this final report in the fact that it deals with the thermodynamic properties of one of the most common and technically important gas mixtures, namely, moist air.

It is deeply to be regretted that this report must record the death late in 1947 of Mr. Walker, Temporary Chairman, International Joint Committee on Psychrometric Data. It was upon his recommendation that Council of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS invited *all organizations likely to be interested in the problem of the properties of mixtures of air and water vapor to appoint a representative on a joint Inter-Society Committee to consider the adoption, when available, of basic equations and tables of such properties.*[1]

Problem

The problem before the International Joint Committee on Psychrometric Data may be regarded as consisting of three parts: (A) to obtain information regarding the thermodynamic properties of moist air which can claim general acceptance as standard on the ground that it is thermodynamically consistent within the accuracy of existing knowledge; (B) to develop specifications of standard instruments for the practical realization of these properties in the field; (C) to recommmend standard symbols, terminology, and charts calculated to promote a better understanding of underlying theory in its application to practical problems involving moist air.

This report sets forth the recommendations of the Working Subcommittee on Part A of the overall problem and explains the reasoning on which these recommendations are based.

Criterion

The Working Subcommittee agreed from the start to try to recommend a formulation, rather than discrete tables, of the thermodynamic properties of moist air, valid over the widest practical ranges of pressure and temperature. While for the majority of present day applications in the field of air conditioning it might suffice to recommend a table of properties at standard atmospheric pressure, such a table would by no means satisfy the needs of the meteorologist or the aerodynamic engineer; moreover, it would be shortsighted to assume that, even in the field of air conditioning engineering, the number of applications in which

†This report was accepted by the International Joint Committee on Psychrometric Data at its meeting on May 12, 1949. Chairman of the Joint Committee is Dr. Baldwin M. Woods, past president of the ASHVE and director of University Extension, University of California, Berkeley.

*Dean, Towne Scientific School, University of Pennsylvania. The other members of the Working Subcommittee are: C. S. Cragoe, National Bureau of Standards; L. P. Harrison, U.S. Weather Bureau; F. G. Keyes, Massachusetts Institute of Technology. S. Gratch, Towne Scientific School, University of Pennsylvania, rendered invaluable aid to the Working Subcommittee as technical adviser.

[1]Exponent numerals refer to Bibliography.

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accurate properties at other pressures are needed will not increase substantially as time goes on. Fortunately, the pressure range of principal interest now and probably for some time in the future is the low-pressure range where well-tested theory is able to make reliable predictions regarding the effect of pressure on the various thermodynamic properties of moist air.

The Working Subcommittee found no difficulty in adopting *thermodynamic consistency within the accuracy of existing knowledge* as the principal criterion by which to judge the validity of its recommendations. The adoption of this criterion at once implies the abrogation of Dalton's Law on which most predictions of the thermodynamic properties of moist air have heretofore been based. By present day standards Dalton's Law must be regarded as an inaccurate conjecture based upon an unwarranted faith in the *ultimate simplicity of nature*. Granted that the law and the consequences inferred more or less intuitively from it are extremely useful as rough and sometimes even close approximations to the truth, their use as a basis for standardization is now completely out of the question.

Dalton's Law

Dalton's Law is a difficult thing to discuss in any clear and unequivocal manner because such a variety of interpretations have come to be placed upon it. If the statement of it given by Preston[2] in his classical treatise, *The Theory of Heat*, be accepted it would appear that the pressure is the only property of a gas mixture which the law attempts to predict; and this, it says, is the sum of the pressures which the constituents would individually exert if each occupied separately the volume of the mixture at the temperature of the mixture. Now statistical mechanics predicts that the law would hold identically if each constituent were a hypothetical perfect gas showing no effects of intermolecular forces whereupon (*a*) the individual pressures would be the individual mol-fractions times the pressure of the mixture, (*b*) the sum of the individual enthalpies would equal the enthalpy of the mixture, and (*c*) the sum of the individual entropies would equal the mixture entropy.

Unfortunately, the preceding predictions (*a*), (*b*), and (*c*), which are strictly valid only in the case of a hypothetical mixture of perfect gases, have been used as vehicles for the introduction of various concepts that cannot be applied without ambiguity or thermodynamic inconsistency to mixtures of actual imperfect gases. It may be stated that Dalton's Law regarding the additivity of individual pressures and its alleged consequences regarding the additivity of individual enthalpies and entropies fails mainly, if not entirely, because of the effects of intermolecular forces between *unlike* molecules while the alleged consequences that the individual pressures are the individual mol-fractions times the pressure of the mixture fail because of the effects of forces between *like* molecules also.

Various modifications of Dalton's Law have been proposed from time to time including the so-called Gibbs-Dalton, Amagat, and Lewis-Randall rules. None of these, however, offers any substantial improvement over Dalton's Law itself; all of them must now be regarded as conjectures based pretty much on wishful thinking.

Statistical Mechanics

Developments in statistical mechanics have given us a well-tested theory of the thermodynamic properties of gas mixtures which is entirely workable in the range of sufficiently low pressures. This theory can be expressed succinctly in the following form,

$$\mu_i = g^\circ_i(T) + RT\log_e(x_i p) + p\,[\Sigma x_i x_k A_{ik}(T) - 2\Sigma x_k A_{ik}(T)] + \ldots\ldots\ldots\ldots (1)$$

where: the $g^\circ_i(T)$ are pure temperature functions called *zero-pressure reduced free enthalpies* which are directly calculable from a knowledge of the spectroscopic constants of the atomic or molecular species in question; the x_i and x_k are mol-fractions; the $A_{ik}(T) = A_{ki}(T)$ are pure temperature functions called *second virial coefficients* ($i = k$) or *interaction coefficients* ($i \neq k$) in terms of which the effects of intermolecular forces are expressed. The *chemical potentials* μ_i, when expressed as functions of temperature, pressure, and the various mol-fractions, are characteristic functions in the sense that from a knowledge of them can be derived all thermodynamic properties of the homogeneous phase to which they belong by application of such identical relations as the following:

$$g = \Sigma x_i \mu_i;\ v = \partial g/\partial p;\ s = -\partial g/\partial T;$$
$$h = \partial(\tau g)/\partial \tau \ldots\ldots\ldots\ldots\ldots\ldots (2)$$

where g, v, s, h denote specific free enthalpy, specific volume, specific entropy, specific enthalpy, respectively; and where τ denotes reciprocal absolute temperature.

Dry Air

Atmospheric air is a mixture of at least a dozen different gases including: oxygen (O_2), nitrogen (N_2), argon (A), carbon dioxide (CO_2)—neon (Ne), helium (He), krypton (Kr), hydrogen (H_2), xenon (Xe), ozone (O_3), radon (Rn)—water vapor (H_2O). Those in the middle group comprise what is commonly referred to as the residual part of atmospheric air. Paneth[3] finds their individual concentrations to be about as follows:

	Percent
Neon	0.0018
Helium	0.00053
Krypton	0.00010
Hydrogen	0.00005
Xenon	0.000008
Ozone (increasing with altitude)	0.000001
Radon (decreasing with altitude)	0.6×10^{-18}

Accordingly the total concentration of the residual part of atmospheric air should be expected not to exceed 0.0025 percent.

The gases in the first group comprise what is commonly called *dry air*. It is true that the relative composition of this group is subject to

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secular variations and variations with location, meteorological conditions, and season, but these variations are produced mainly, if not entirely, by variations in the concentration of carbon dioxide. All evidence indicates that the average concentration of carbon dioxide in dry air in the troposphere is very close to 0.03 percent. Carpenter[4] recommends the value 0.031 percent.

For the purpose of standardization the composition of dry air can be defined more or less arbitrarily. It would seem desirable, however, to base the definition on available information regarding the average composition of atmospheric air in the troposphere. Carpenter[4] finds that dry air contains 20.939 percent oxygen and 0.031 percent carbon dioxide. Paneth[3] accepts the value, 20.95 percent oxygen in dry air containing carbon dioxide, recommended by Benedict[5]. The National Bureau of Standards has come to prefer the values, 20.94 percent oxygen and 0.03 percent carbon dioxide, which are the values originally agreed upon by the Working Subcommittee. It later developed, however, that the International Meteorological Organization holds a strong preference for the Paneth[3] values quoted previously. Hence, and particularly in view of the fact that the definition of what is to be referred to as *dry air* must be decided upon more or less arbitrarily, the writer agreed on behalf of his Working Subcommittee to accept the values, 20.95 percent oxygen and 0.03 percent carbon dioxide.

Cady and Cady[6] find that dry air free of CO_2 contains 0.93 percent argon by volume. This figure would not have to be modified appreciably to apply to dry air containing 0.03 percent carbon dioxide; hence it was accepted by the Working Subcommittee.

The (revised) recommendation of the Working Subcommittee regarding the composition of dry air is as follows:

Oxygen (O_2)	0.2095
Nitrogen (N_2)	0.7809
Argon (A)	0.0093
Carbon Dioxide (CO_2)	0.0003
	1.0000

This is a *fixed, mol-fraction* composition which the Subcommittee intends should be regarded as *exact by definition*. The fact that the foregoing figures were derived from experimental determinations made with an Orsat-like apparatus which does not yield information regarding the mol-fractions directly was disregarded because it is almost certain that the experimental data themselves are not sufficiently precise to justify any attempt at refinement; moreover, as has been stated previously, any definition of what is to be called *dry air* necessarily involves some exercise of arbitrariness.

The definition of dry air adopted by the International Meteorological Organization differs slightly from that recommended in the preceding paragraph, by including the Paneth[3] values of the concentrations (converted to mol-fractions) of the residual gases. This slight difference will not affect the calculated values of the various thermodynamic properties of dry air, at any rate, not within the accuracy of existing knowledge.

The Working Subcommittee recommends adoption of the International Committee on Atomic Weights (1947) values of the relevant molecular weights[7], namely,

Oxygen (O_2)	32.0000
Nitrogen (N_2)	28.016
Argon (A)	39.944
Carbon Dioxide (CO_2)	44.01
Water (H_2O)	18.016

These yield 28.966, for the apparent molecular weight of dry air.

Physical Constants

The Working Subcommittee recommends that each individual zero-pressure reduced free enthalpy $g^\circ_i(T)$ appearing in Equation 1 be computed from best available spectroscopic data on the energy levels of the molecule in question, by the methods of statistical mechanics. Certain fundamental physical constants which have to be determined experimentally and which are therefore subject to experimental error enter these computations. These include: Planck's Constant h, Boltzmann's Constant k, speed of light c, Avogadro's Number N_o, and the relevant molecular weights M_i. For the first four of these the Subcommittee recommends adoption of the Birge[8] (1941) values as follows:

$h = (6.6242 \pm 0.0024) \times 10^{-27}$ .....erg sec

$k = (1.380474 \pm 0.00026) \times 10^{-16}$ .....erg/°K

$c = (2.99776 \pm 0.00004) \times 10^{10}$ .....cm/sec

$N_o = (6.02283 \pm 0.0011) \times 10^{23}$ .....gmol^{-1}

DuMond and Cohen[9] have treated the various input data from which the values of the above constants have to be computed in a more impartial manner than Birge was able to employ in 1941. Theirs may shortly come to supersede the Birge (1941) values. Fortunately, they leave the combinations kN_o, hcN_o, and hc/k unchanged in value. This means that the only effects of using the DuMond-Cohen (1948) instead of the Birge (1941) values in cal-

Note: Place rightmost digit of tolerance beneath rightmost digit of value to which it applies, regardless of decimal point location.

Table 1—Zero-Pressure Properties of Dry Air

t °C	h°_a (kcal/kg)	Tol. ±	s°_a (kcal/kg°K)	Tol. ±
-90	43.771	18	1.52160	61
-80	46.165	18	1.53431	61
-70	48.560	19	1.54640	62
-60	50.953	20	1.55791	62
-50	53.348	21	1.56889	63
-40	55.743	22	1.57938	63
-30	58.138	23	1.58944	64
-20	60.533	24	1.59910	64
-10	62.929	25	1.60838	64
0	65.326	26	1.61732	65
10	67.724	27	1.62594	65
20	70.122	28	1.63427	65
30	72.523	29	1.64231	66
40	74.924	30	1.65010	66
50	77.326	31	1.65765	66
60	79.730	32	1.66498	67
70	82.135	33	1.67210	67
80	84.542	34	1.67901	67
90	86.952	35	1.68574	67

Table 2—Zero-Pressure Properties of Water Vapor

t °C	h°_w (kcal/kg)	Tol. ±	s°_w (kcal/kg°K)	Tol. ±
-90	80.290	24	2.28627	69
-80	84.709	25	2.30975	69
-70	89.130	27	2.33207	70
-60	93.552	28	2.35331	71
-50	97.975	29	2.37359	71
-40	102.400	31	2.39299	72
-30	106.828	32	2.41158	72
-20	111.259	33	2.42945	73
-10	115.693	35	2.44661	73
0	120.130	36	2.46317	74
10	124.573	37	2.47914	74
20	129.020	39	2.49459	75
30	133.473	40	2.50951	75
40	137.932	41	2.52399	76
50	142.399	43	2.53803	76
60	146.873	44	2.55166	77
70	151.355	45	2.56492	77
80	155.845	47	2.57780	77
90	160.345	48	2.59037	78

culating the thermodynamic properties of moist air would be (*a*) to reduce the estimated uncertainties of the results and (*b*) to shift by a constant amount the so-called *absolute* entropy of each constituent. At present, however, the air conditioning engineer, the meteorologist, and the aerodynamicist are interested in *relative* values only of enthalpy and entropy; hence effect (*b*) is of no practical consequence at present.

Conversion Factors

The Working Subcommittee also recommends adoption of the following conversion factors:

1.8	°R/°K
30.48	cm/ft
453.5924	g/lb
1	Btu • kg • °K/kcal • lb • °R
860/1.00019	kcal/abs • kwhr
1,013,250	dyne/cm² • atm

The last two of these will bear some explanation. The symbol kcal is intended to designate the so-called *steam tables kilocalorie* defined in the report of the First International Steam Tables Conference[10] as 1/860 (mean) international kilowatt-hour. Birge (1941) recommends the conversion factor 1.00020 ± 0.000045 (mean) international kilowatt-hour per absolute kilowatt-hour. The International Committee of Weights and Measures, at its meetings in October 1946, revised the value of this factor downward to 1.00019. This revision should, of course, be accepted especially since none of the other fundamental constants or conversion factors listed is affected thereby.

According to Birge (1941) the mean density of the mercury in a column 76 cm high at 0 C and 980.665 cm/sec² is 13.59504 ± 0.000057 g/cm³. If the standard atmosphere be defined as the pressure exerted by such a column it would be equal to 1,013,246 ± 4 dyne/cm². This is the definition adopted by the International Meteorological Organization. Alternatively, the standard atmosphere is defined in some quarters as equal to 1,013,250 dyne/cm², exactly. This is the definition recommended by the Working Subcommittee. The difference between the two atmospheres is exceedingly small but is mentioned here for the sake of definiteness.

Zero-Pressure Reduced Free Enthalpies

By way of introduction it may be well to mention that, in accordance with relations in Equation 2, zero-pressure enthalpy h° and zero-pressure reduced entropy s° (by reduced entropy is meant the sum, $s + R\ln p$) are derivable from zero-pressure reduced free enthalpy g° (by reduced free enthalpy is neant the difference, $g - RT\ln p$) as follows:

$$h^\circ = d(\tau g^\circ)/d\tau \; ; \; s^\circ = -dg^\circ/dT$$

Oxygen: Gratch has reduced the recently computed data of Woolley[11] to the following semi-rational equation:

$$g^\circ/RT = (7/2)\ln\tau + \ln(1-e^{-\theta\tau}) - A\tau - B\tau^2 - D/\tau - F/(e^{-\theta\tau}-1) - 1.2164$$

with

$$\theta = 2235.4\ ^\circ K$$
$$A = 1.073\ ^\circ K$$
$$B = 0$$
$$D = 3.30 \times 10^{-6}\ ^\circ K^{-1}$$
$$F = 0.0111$$

Values of zero-pressure enthalpy h° derived from this equation agree with the Woolley data to within a maximum deviation of 0.010 percent and a root mean square deviation of 0.005 percent in the range 100 to 600°K. Values of zero-pressure reduced free enthalpy g° and zero-pressure reduced entropy s° show maximum deviations of 0.002 percent and root mean square deviations of 0.001 percent from the Woolley data in this range. These data themselves are probably accurate to within 0.015 percent in the range 100 to 600°K so that a conservative estimate of the probable error of the above equation in this range is 0.02 percent in h°, g°, and s°.

Nitrogen: Gratch[12] has reduced his own calculations of the zero-pressure properties of nitrogen to the following semi-rational equation:

$$g^\circ/RT = (7/2)\ln\tau + \ln(1-e^{-\theta\tau}) - A\tau - B\tau^2 - D/\tau - F/(e^{-\theta\tau}-1) + 0.414686$$

with

$$\theta = 3352.69\ ^\circ K$$
$$A = 0.9580\ ^\circ K$$
$$B = 0.09\ ^\circ K^2$$
$$D = 2.023 \times 10^{-6}\ ^\circ K^{-1}$$
$$F = 0.009009$$

The estimated uncertainty in values of g° calculated from this equation is 0.011 percent in the range 50 to 800°K.

Argon: Gratch and Van Voorhis[13] give the following equation valid in the range 50 to 2000°K with a probable error in g° itself of about 0.02 percent:

$$g^\circ/RT = (5/2)\ln\tau - 1.867.$$

Carbon Dioxide: Gratch has reduced the tabulated data of Wagman et al[14] to the following semi-rational equation:

Table 3—Second Virial Coefficients of Dry Air

t °C	A_{aa} (cm³/gmol)	Tol. ±	B_{aa} (cm³/gmol)	Tol. ±	C_{aa} (cm³/gmol °K)	Tol. ±
-90	50.4	39	180.6	195	-0.711	107
-80	43.9	33	161.7	167	-0.610	86
-70	38.2	29	145.7	143	-0.529	71
-60	33.2	25	131.9	124	-0.463	58
-50	28.9	22	120.0	108	-0.409	48
-40	25.0	19	109.7	95	-0.363	41
-30	21.6	17	100.6	83	-0.325	34
-20	18.5	15	92.5	74	-0.292	29
-10	15.7	13	85.3	66	-0.265	25
0	13.2	12	78.9	59	-0.240	22
10	10.9	11	73.1	53	-0.220	19
20	8.81	95	67.8	48	-0.201	16
30	6.87	86	63.1	43	-0.1853	142
40	5.09	78	58.7	39	-0.1712	125
50	3.44	71	54.7	36	-0.1586	110
60	1.91	65	51.0	32	-0.1474	98
70	0.49	59	47.6	30	-0.1373	87
80	-0.84	54	44.5	27	-0.1283	77
90	-2.07	54	41.6	25	-0.1201	69

Table 4—Second Virial Coefficients of Water Vapor

t °C	A_{vv} (cm³/gmol)	Tol. ±	B_{vv} (cm³/gmol)	Tol. ±	C_{vv} (cm³/gmol °K)	Tol. ±
-90	-	-	-	-	-	-
-80	-	-	-	-	-	-
-70	-	-	-	-	-	-
-60	-	-	-	-	-	-
-50	-	-	-	-	-	-
-40	-	-	-	-	-	-
-30	-	-	29000	26000	-100	110
-20	-	-	20800	9500	- 71	38
-10	2300	1900	15600	3800	- 51	14
0	1830	800	12000	1600	- 37.2	59
10	1510	400	9440	800	- 28.0	28
20	1260	210	7570	420	- 21.5	14
30	1074	116	6180	230	- 16.84	76
40	924	66	5120	130	- 13.40	42
50	803	40	4298	80	- 10.81	25
60	705	25	3653	49	- 8.85	15
70	625	16	3138	31	- 7.324	90
80	558	10	2722	21	- 6.129	59
90	501	7	2382	14	- 5.179	39

Table 5—Third Virial Coefficients of Water Vapor

t °C	A_{www} (cm³/gmol atm)	Tol. ±	B_{www} (cm³/gmol atm)	Tol. ±	C_{www} (cm³/gmol atm°K)	Tol. ±
0	-	-	-	-	-	-
10	-	-	-	-	-	-
20	-	-	15000	18000	-47	61
30	-	-	8100	5300	-25	17
40	-	-	4600	1800	-14.0	58
50	-	-	2770	640	- 8.0	20
60	110	120	1710	240	- 4.80	72
70	72	50	1090	100	- 2.97	29
80	48	21	709	43	- 1.87	12
90	33	10	474	20	- 1.21	6

Table 6—Interaction Coefficients of Moist Air

t °C	A_{aw} (cm³/gmol)	Tol. ±	B_{aw} (cm³/gmol)	Tol. ±	C_{aw} (cm³/gmol °K)	Tol. ±
-90	84.2	86	226	17	-0.774	93
-80	77.0	82	207	16	-0.674	81
-70	70.7	78	191	15	-0.591	71
-60	65.1	75	177	14	-0.523	63
-50	60.2	72	164	13	-0.466	56
-40	55.8	70	153	13	-0.418	50
-30	51.8	67	143	12	-0.376	45
-20	48.2	65	135	12	-0.341	41
-10	45.0	63	126.6	112	-0.310	37
0	42.0	61	119.5	108	-0.284	34
10	39.3	60	113.0	104	-0.260	31
20	36.8	58	107.0	100	-0.240	29
30	34.5	57	101.6	97	-0.221	27
40	32.3	55	96.5	94	-0.205	25
50	30.4	54	91.9	91	-0.190	23
60	28.5	53	87.6	89	-0.177	21
70	26.8	52	83.6	86	-0.165	20
80	25.2	51	79.9	84	-0.155	19
90	23.7	50	76.4	82	-0.145	17

$$g^\circ/RT = (7/2)\ln\tau + 2\ln(1-e^{-\theta_1\tau}) + \ln(1-e^{-\theta_2\tau}) + \ln(1-e^{-\theta_3\tau}) - 1.8945$$

with

$\theta_1 = 960$ °K
$\theta_2 = 1944$ °K
$\theta_3 = 3379$ °K

In the range 300 to 600°K, values of h° and s° derived from this equation agree with the Wagman data within 0.05 percent and 0.005 percent respectively. The probable error of the equation itself is thought to be less than 0.1 percent in this range.

Water Vapor: Goff and Gratch[15] give what amounts to the following semi-rational equation:

$$g^\circ/RT = 4\ln\tau + \ln(1-e^{-\theta_1\tau}) + \ln(1-e^{-\theta_2\tau}) + \ln(1-e^{-\theta_3\tau}) - a_1e^{-\theta_1\tau} - a_2e^{-\theta_2\tau} - a_3e^{-\theta_3\tau} - A\tau - D/\tau + \text{const.}$$

with

$\theta_1 = 2291.16$ °K
$\theta_2 = 5176.37$ °K
$\theta_3 = 5445.59$ °K
$a_1 = -0.03958$
$a_2 = +0.05353$
$a_3 = +0.04000$
$A = 5.011$ °K
$D = 2.32 \times 10^{-5}$ °K^{-1}

Gratch has subsequently adjusted the constant by comparison with the Wagman[14] data as follows:

const. = 4.1083

The probable error of values of g° calculated from the equation is 0.015 percent in the range —90 to +90°C.

Zero-Pressure Properties Of Dry Air

All necessary data are now at hand to compute values of the zero-pressure enthalpy h°_a and zero-pressure reduced entropy s°_a for dry air in accordance with the relations,

$$h^\circ_a = \Sigma x_i h^\circ_i \text{ and } s^\circ_a = \Sigma x_i s^\circ_i \quad \ldots\ldots (3)$$

The numerical value of the gas constant R is obtained by multiplying Boltzmann's Constant k into Avogadro's Number N_o and applying appropriate conversion factors from the approved list given earlier in this report; the result is

$$R = 1.98583 \pm 0.00022 \text{ kcal/kmol °K.}$$

The apparent molecular weight of dry air is

$$M_a = 28.966 \pm 0.0016 \text{ kg/kmol,}$$

the indicated uncertainty arising solely from uncertainties in the molecular weights of its several constituents.

In Table 1 are listed the values so computed, at 10-deg intervals of Centigrade temperature t(°C) defined in terms of absolute temperature T(°K) by the relation,

$$T(\text{°K}) = t(\text{°C}) + 273.16 \quad \ldots\ldots (4)$$

Recommended tolerances set at 0.04 percent of the values to which they apply are also listed in Table 1.

Zero-Pressure Properties of Water Vapor

The necessary data are also at hand to compute the zero-pressure enthalpy h°_w and zero-pressure reduced entropy s°_w of water vapor. The present best value of the molecular weight of water is

$$M_w = 18.016 \pm 0.0002 \text{ kg/kmol.}$$

Values of these two properties are listed in Table 2 together with recommended tolerances set at 0.03 percent.

Virial Coefficients of Dry Air

The temperature function $A_{aa}(T)$ is called the *second virial coefficient* of dry air. From Bridgeman's[16] adjustment of the constants in the well-known Beattie-Bridgeman equation of state, its dependence upon absolute temperature T(°K) is given by

$$A_{aa} = -40.70 + 13116\tau + 12 \times 10^7\tau^3 \quad \text{cm}^3/\text{gmol}$$

Table 3 lists values of A_{aa} and of the derived coefficients,

$B_{aa} = d(\tau A_{aa})/d\tau$ and
$C_{aa} = dA_{aa}/dT$,

together with recommended tolerances obtained by comparison with other formulations.

Strictly speaking the values listed in Table 3 apply to dry, CO_2-free air. The Subcommittee, however, recommends them as applying to dry air as defined in this report because it estimates that the corrections necessary to achieve complete consistency would be well within the probable error of the values themselves.

Third Virial Coefficients

Following the rightmost term in Equation 1 comes a term proportional to the square of the pressure (p^2) in which the so-called *third virial coefficients* $A_{ijk}(T)$ appear. By analogy, this term is $+\frac{1}{2}p^2(2\Sigma x_i x_j x_k A_{ijk} - 3\Sigma x_i x_k A_{ijk})$, the value of A_{ijk} being unchanged by permutation of the indices i,j,k. The third virial coefficient of dry air $A_{aaa}(T)$ is entirely negligible in the range 150 to 400°K, 0 to 10 atm (atmosphere).

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Virial Coefficients of Water Vapor

Goff and Gratch[17] have formulated the data on the second and third virial coefficients of water vapor as follows:

$$A_{ww} = -33.97 + 55306\tau \cdot 10^{72000\tau^2} \text{ cm}^3/\text{gmol}$$

$$A_{www} = 0.0348\tau^2 A^3_{ww} \text{ cm}^3/\text{gmol} \cdot \text{atm}$$

Values of these and the corresponding derived coefficients, $B = d(\tau A)/d\tau$ and $C = dA/dT$, are listed in Table 4 and Table 5 together with tolerances set at twice the probable error of the values themselves as inferred from the deviations in the experimental data on which they are based. Higher virial coefficients make negligible contributions at pressures below the saturation pressure up to at least 400°K.

Interaction Coefficients of Moist Air

For the purpose of thermodynamic analysis moist air may be regarded as a binary mixture of dry air and water vapor. By this simplification the number of chemical potentials μ_i is reduced to two, namely μ_a and μ_w. Each of these is given as to form by Equation 1; each contains, besides the second virial coefficients A_{aa} and A_{ww}, an interaction coefficient A_{aw}. Goff and Gratch[15] have formulated data obtained under a cooperative investigation between the Towne Scientific School, University of Pennsylvania, and the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS into the following semi-rational equation:

$$A_{aw} = -29.53 + 0.00669T(1-e^{-\theta\tau}) + A\tau + B\tau^2 + D\tau^3 \text{ cm}^3/\text{gmol}$$

with

$\theta = 4416.5$ °K
$A = 17{,}546$ cm³ °K/gmol
$B = 95{,}300$ cm³ °K²/gmol
$D = 8.515 \times 10^7$ cm³ °K³/gmol

Values of A_{aw} and of the corresponding derived coefficients, B_{aw} and C_{aw}, are listed in Table 6 together with tolerances set at thrice the estimated probable errors of the values to which they refer. These probable errors are difficult to estimate reliably because of the meagerness of experimental data; hence they are multiplied by three instead of two to get the corresponding tolerances.

Each chemical potential also contains, besides the third virial coefficients A_{aaa} and A_{www}, the higher order interaction coefficients A_{aaw} and A_{aww}. Pending further research it is necessary to assume that of these four coefficients only A_{www} makes any appreciable contribution to the thermodynamic properties of moist air, at any rate, in the range 150 to 400°K and 0 to 3 atm.

Saturation Pressure of Water

Goff and Gratch[17] have calculated the saturation pressure of pure water from other relevant data by means of the identical relations of thermodynamics. Their calculations involve an integration, over the logarithm of absolute temperature, of an integrand which varies only slowly with temperature. Their results may therefore be regarded as determining the function $p_s(T)$ or the function $p_s(t)$, where t is the Centigrade temperature defined by Equation 4.

a. Saturation Pressure of Ice, —100 to 0°C. Values of the saturation pressure of pure ordinary water ice are to be computed from the Goff-Gratch formula,

$$\log_{10} p_{si} = -9.09718(T_o/T - 1) - 3.56654 \log_{10}(T_o/T) + 0.876793(1 - T/T_o) + \log_{10} p_{si}(T_o) \quad \ldots\ldots\ldots\ldots\ldots\ldots (5)$$

with $T_o = 273.16$°K = 491.688°R and with $p_{si}(T_o) = 0.0060273$ atmospheres, a constant integration which has been adjusted so as to make p_{si} equal to p_{sf} (see following paragraphs) at $T = 273.17$°K, the triple point of water.

The formula as published[17] gives the logarithm of the saturation pressure as the sum of three terms only one of which is explicit in temperature, the other two having been left in the form of definite integrals, over the logarithm of absolute temperature, whose integrands contain both temperature and saturation pressure. In computing numerical values it is easy to evaluate these integrals by the method of successive approximations. For the greater convenience of the user, however, Gratch[18], has replaced these integrals by empirical expressions whose constants he has then adjusted by the method of least squares to make the formula (Equation 5) yield values of $\log_{10}[p_{si}/p_{si}(T_o)]$ showing an average deviation of 0.002 percent and a maximum deviation of 0.015 percent from those given by the formula as previously published.

b. Saturation Pressure of Liquid Water, —50 to 100°C. Values of the saturation

Table 7—Saturation Pressure of Pure Water (millibars[a])

t °C	p_{sf}	Tol. ±	p_{si}	Tol. ±
-90	-	-	0.0000967	9
-80	-	-	0.0005472	44
-70	-	-	0.002615	19
-60	-	-	0.01080	7
-50	0.06355	-	0.03935	22
-40	0.18909	-	0.1283	6
-30	0.50888	-	0.3799	14
-20	1.2543	-	1.032	3
-10	2.863	-	2.597	5
0	6.1078	59	6.107	6
10	12.272	10	-	-
20	23.373	20	-	-
30	42.430	29	-	-
40	73.777	37	-	-
50	123.40	6	-	-
60	199.26	10	-	-
70	311.69	16	-	-
80	473.66	24	-	-
90	701.14	35	-	-

[a] 1 millibar = 1000 dyne/cm²

Table 8—The Coefficient $f_s(p,T)$

	p = 300 mb.[a]		p = 700 mb.		p = 1100 mb.	
t °C	f_{sf}	f_{si}	f_{sf}	f_{si}	f_{sf}	f_{si}
-90	-	1.0027	-	-	-	-
-80	-	1.0024	-	1.0057	-	1.0089
-70	-	1.0022	-	1.0051	-	1.0080
-60	-	1.0020	-	1.0046	-	1.0073
-50	1.0018	1.0018	1.0042	1.0042	1.0065	1.0066
-40	1.0017	1.0017	1.0038	1.0039	1.0060	1.0061
-30	1.0016	1.0016	1.0036	1.0036	1.0055	1.0056
-20	1.0015	1.0015	1.0034	1.0034	1.0052	1.0052
-10	1.0015	1.0015	1.0032	1.0033	1.0049	1.0050
0	1.0016	1.0016	1.0032	1.0032	1.0047	1.0048
10	1.0018	-	1.0032	-	1.0047	-
20	1.0020	-	1.0034	-	1.0048	-
30	1.0023	-	1.0037	-	1.0050	-
40	1.0026	-	1.0041	-	1.0054	-
50	-	-	1.0045	-	1.0059	-
60	-	-	1.0048	-	1.0064	-

[a] 1 millibar = 1000 dyne/cm²

pressure of pure ordinary liquid water are to be computed directly from the Goff-Gratch formula[17], namely,

$$\log_{10} p_{st} = -7.90298\,(T_s/T - 1) + 5.02808\,\log_{10}(T_s/T) - 1.3816 \times 10^{-7}\,(10^{11.344(1 - T/T_s)} - 1) + 8.1328 \times 10^{-3}\,(10^{-3.49149(T_s/T - 1)} - 1) + \log_{10} p_{st}(T_s) \quad (6)$$

with $T_s = 373.16°K = 671.688°R$ and with $p_{st}(T_s) = 1$ atm.

c. Undercooled Liquid: At temperatures below 0°C it is not possible for the vapor and liquid phases of pure water to coexist in stable equilibrium; it is an experimental fact, however, that they can coexist in what is called *metastable* equilibrium, in which case the coexisting liquid phase is called *undercooled liquid.*

Since reliable experimental data on the thermodynamic properties of undercooled liquid are not presently available, interim values can only be obtained by some sort of extrapolation. Washburn[19] has, in effect, extrapolated the specific heat data of Barnes and Cooke[20] in the range, —5 to 5°C, to develop a formula for $\log_{10}(p_{st}/p_{s1})$ which has heretofore been widely used. This formula may now be corrected to make it consistent with the Osborne[21] value of the heat of fusion of ice at 0°C and with the Giauque and Stout[22] measurements of the specific heat of ice.

Gratch[23] has calculated values of the ratio p_{st}/p_{s1} from: (*a*) the original Washburn formula; (*b*) the corrected Washburn formula; (*c*) the Osborne, Stimson, and Ginnings[24] formulation of the calorimetric quantity γ using the approximate relation $\gamma = RT^2 d\ln p_{st}/dT$; (*d*) the corrected Washburn formula adjusted to fit the Scheel and Heuse[25] experimental measurements of p_{st} down to about —15°C after omitting the last term which is quite small in the experimental range but increases very rapidly as temperature decreases; and (*e*) the Goff-Gratch formulas quoted. The results of these calculations are compared as follows:

Saturation Pressure of Undercooled Liquid

	(p_{st}/p_{s1})				
t(°C)	(*e*)	(*a*)-(*e*)	(*b*)-(*e*)	(*c*)-(*e*)	(*d*)-(*e*)
0	1.0001	—.0001	0.0000	0.0000	0.0000
—10	1.1023	—.0002	—.0001	+.0005	—.0014
—20	1.2154	—.0004	—.0003	+.0013	—.0018
—30	1.3395	—.0014	—.0009	+.0031	—.0933
—40	1.4738	—.0039	—.0031	+.0072	—.0920
—50	1.6151	—.0077	—.0098	+.0167	+.0044

Another comparison based on an extrapolation of the Smith and Keyes[26] data on the specific volume v_f of the saturated liquid is of interest. Using the Goff-Gratch values of p_{st} together with previously quoted information regarding zero-pressure enthalpy $h°_w$, the virial coefficients A_{ww}, A_{www}, and the corresponding derived coefficients B_{ww}, B_{www}, we may compute values of the specific enthalpy h_f of the saturated liquid from the identical relation, $h_f = h_g - (v_g - v_f)(dp_{st}/dT)$. It is safe to ignore the extremely small variation with temperature of the difference between h_f and the specific enthalpy h'_w of compressed liquid at atmospheric pressure and regard values of the quantity dh_f/dT as values of the isobaric specific heat c'_w at atmospheric pressure. In the following table these are compared with (*a*) values obtained from extrapolation of the Osborne, Stimson, and Ginnings[24] formula and (*b*) values obtained from extrapolation of the Washburn[19] formulation of the Barnes and Cooke[20] data.

Specific Heat of Undercooled Liquid

t(°C)	Goff-Gratch	NBS	Barnes and Cooke
—10	1.018	1.022	1.023
—20	1.039	1.055	1.045
—30	1.075	1.130	1.074
—40	1.14	1.30	1.11
—50	1.27	1.70	1.15

The saturation pressure of undercooled liquid is of such great concern to the meteorologist, if not to the engineer, that it seems necessary to recommend one of these extrapolations pending further research. The International Meteorological Organization recommends extrapolation of the Goff-Gratch formula down to —50°C because it insures continuity of 0°C and minimizes the number of formulas to be quoted.

Table 7 lists values of saturation pressure calculated from the Goff-Gratch formulas. Recently Keyes[27] has reviewed and reformulated existing data on this important property in the range 0 to 150°C in the light of the findings of Blaisdell and Kaye[28] regarding the relation between the absolute and the International Centigrade temperature scales. His results are in remarkably close agreement with the values listed in Table 7 for temperatures above 0°C.

Regarding the tolerances listed in Table 7, each of those below 0°C has been set at thrice the estimated probable error of the corresponding saturation pressure; those above 0°C are the ones recommended by the Third International Conference on Steam Tables[29]; no attempt has been made to set tolerances on the recommended values of the saturation pressure of undercooled liquid.

Above 0°C the recommended values of saturation pressure differ slightly, but nevertheless significantly, from those previously recommended by the Third International Conference on Steam Tables[29] if the difference between the absolute and the International Centigrade temperature scales be ignored in making the comparison. The analysis of Keyes[27] may be taken to show, however, that the two sets are brought into remarkably close agreement by transferring the latter from the International Centigrade to the absolute scale by means of the Blaisdell and Kaye[28] findings regarding the relation between the two scales. It seems clear, therefore, that the tolerances recommended by the Third International Conference on Steam Tables could stand reduction.

Saturation of Moist Air

Moist air is said to be saturated when its condition is such that it can coexist in neutral equilibrium with an associated condensed phase presenting a flat surface to it. Strictly speaking, the associated condensed phase is not pure water, but water containing small concentrations of dissolved gases. The relative concentrations of these dissolved gases are not the same as those of the same gases in the vapor phase, but we are practically compelled to ignore the very slight differences and to regard the condensed phase as an ideal mixture of pure water and dry air having but two component chemical potentials, μ'_w and μ'_a.

In line with the foregoing simplification we can say that moist air is saturated when it has the same values of temperature, pressure, and its two component chemical potentials as does the associated condensed phase. There are thus four relations to be satisfied at saturation. These four relations involve, however, six independent variables, namely, the two temperatures, the two pressures, and the two mol-fractions; hence, the two-phase system has two degrees of freedom at saturation. This means that such quantities as the weight of water, the volume, the enthalpy, and the entropy—all per unit weight of dry air—must be re-

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garded as functions of two independent variables which are usually chosen to be pressure p and temperature T.

Humidity Ratio at Saturation

The weight of water vapor per unit weight of dry air is called *humidity ratio* or *mixing ratio*. It is denoted by the letter W to which may be affixed the subscript $_s$ to specify its value at saturation. It is convenient to introduce a coefficient $f_s(p,T)$ defined by

$$W_s = 0.62197 \frac{f_s p_s/p}{1 - f_s p_s/p} \quad \ldots\ldots (7)$$

where p_s is the saturation pressure of pure water, a function of temperature only, and $0.62197 = 18.0160/28.966$.

Goff and Gratch[30,31] have reduced the conditions for saturation in the case of moist air to

$$\log f_s = \alpha(1 - p_s/p) + \beta(p/p_s - 1) \quad \ldots\ldots (8)$$

with

$$\alpha = (A_{aa} - 2A_{aw} + A_{ww})p_s/RT + A_{www}p_s^2/RT$$

$$\beta = -k'p_s + v'_w p_s/RT + (A_{aa} + 2A_{aw})p_s/RT$$

where

k' = solubility coefficient of air in water,

v'_w = specific volume of (liquid or solid) water.

Table 8 gives an idea of how the coefficient $f_s(p,T)$ varies with pressure and temperature. In computing the data listed in Table 8, values of k' for liquid water above 0°C have been taken from *International Critical Tables*[32]; k' has been assumed to be zero for undercooled liquid and ice. Values of v'_w for liquid water above 0°C have been taken from Osborne, Stimson, and Ginnings[33]; a constant value of 18.0 cm³/gmol has been assumed for undercooled liquid; values for ice have been taken from Jakob and Erk[34] and from Keenan and Keyes[35].

Rather than list the values of k' and v'_w used in calculating values of the coefficient $f_s(p,T)$ it seems better to list, as is done in Table 9, values of the coefficients α and β in which they appear along with other properties discussed previously. In Table 9 the tolerances given are based on the previously stated tolerances for A_{aa}, A_{aw}, etc., on an arbitrarily chosen tolerance of 10 percent in k', and on zero tolerance in v'_w.

Table 10 lists values of humidity ratio at saturation at standard atmospheric pressure together with tolerances at least twice estimated probable error.

Volume

The correct expression for the specific volume of moist air can be derived from Equation 1 for the chemical potentials by forming the specific free enthalpy g according to the identical relation, $g = \Sigma x_i \mu_i$, and differentiating this according to the identical relation, $v = \partial g/\partial p$. The volume of moist air per unit weight of dry air is obtained from its specific volume in an obvious manner. Table 11 lists values of the volume of moist air per unit weight of dry air at standard atmospheric pressure, together with tolerances at least twice the estimated probable error.

Enthalpy

The correct expression for the specific enthalpy of moist air can be derived from Equation 1 for the chemical potentials by forming the specific free enthalpy g according to the identical relation, $g = \Sigma x_i \mu_i$, and differentiating this according to the identical relation, $h = \partial(\tau g)/\partial\tau$, where τ denotes reciprocal absolute temperature. The enthalpy of moist air per unit weight of dry air is obtained from its specific enthalpy in an obvious manner.

The enthalpy of moist air per unit weight of dry air is subject to augmentation by amount $\overline{h}_a + W\overline{h}_w$, where $\overline{h}_a$ and $\overline{h}_w$ are arbitrary constants. In Table 12 the constant $\overline{h}_a$ has been adjusted so as to make the specific enthalpy of dry air zero at 0°C and standard atmospheric pressure; the constant $\overline{h}_w$ has been adjusted so as to make the specific enthalpy of saturated liquid water zero at 0°C in conformity with the usual steam table practice.

Entropy

The correct expression for the specific entropy of moist air can be derived from Equation 1 for the chemical potentials by forming the specific free enthalpy g according to the identical relation, $g = \Sigma x_i \mu_i$, and differentiating this according to the identical relation,

Table 9—Coefficients in the Expression for Saturation

t °C	α_f	Tol. ±	α_i	Tol. + −	β_f	Tol. + −	β_i	Tol. ±
-90	-	-	0.0000	4	-	-	0.87x10⁻⁹	13
-80	-	-	0.0000	4	-	-	0.44x10⁻⁸	7
-70	-	-	0.0000	4	-	-	1.90x10⁻⁸	29
-60	-	-	0.0000	4	-	-	0.71x10⁻⁷	11
-50	0.0000	-	0.0000	4	3.7 x10⁻⁷	-	2.35x10⁻⁷	35
-40	0.0000	-	0.0000	4	1.02x10⁻⁶	-	0.70x10⁻⁶	11
-30	0.0001	-	0.0001	4	2.52x10⁻⁶	-	1.91x10⁻⁶	29
-20	0.0002	-	0.0001	4	0.57x10⁻⁵	-	0.48x10⁻⁵	7
-10	0.0003	-	0.0003	4	1.21x10⁻⁵	-	1.11x10⁻⁵	17
0	0.00048	42	0.0005	4	2.37x10⁻⁵	36	2.43x10⁻⁵	-
10	0.00076	41	-	-	4.44x10⁻⁵	67	-	-
20	0.00117	41	-	-	0.79x10⁻⁴	12	-	-
30	0.00174	41	-	-	1.34x10⁻⁴	20	-	-
40	0.00251	41	-	-	2.19x10⁻⁴	33	-	-
50	0.00352	42	-	-	3.46x10⁻⁴	52	-	-
60	0.00483	44	-	-	5.26x10⁻⁴	79	-	-
70	0.00649	46	-	-	0.78x10⁻³	12	-	-
80	0.00854	49	-	-	1.12x10⁻³	17	-	-
90	0.01103	56	-	-	1.58x10⁻³	24	-	-

Table 10—Humidity Ratio at Saturation
(Standard Atmospheric Pressure)

t °C	W_{sf}	Tol. ±	W_{si}	Tol. + −
-90	-	-	0.5991x10⁻⁷	66
-80	-	-	0.3386x10⁻⁶	34
-70	-	-	0.1617x10⁻⁵	14
-60	-	-	0.6677x10⁻⁵	53
-50	0.3924 x10⁻⁴	-	0.2430x10⁻⁴	17
-40	1.168 x10⁻⁴	-	0.7923x10⁻⁴	47
-30	0.3141 x10⁻³	-	0.2345x10⁻³	11
-20	0.7744 x10⁻³	-	0.6370x10⁻³	26
-10	1.7688 x10⁻³	-	1.6057x10⁻³	48
0	0.37886x10⁻²	80	3.7886x10⁻³	80
10	0.7659 x10⁻²	15	-	-
20	0.14754x10⁻¹	28	-	-
30	0.27319x10⁻¹	49	-	-
40	0.49114x10⁻¹	79	-	-
50	0.8680 x10⁻¹	13	-	-
60	0.15340	25	-	-
70	0.27884	47	-	-
80	0.55200	105	-	-
90	1.4160	38	-	-

$s = -\partial g/\partial T$. The entropy of moist air per unit weight of dry air is obtained from its specific entropy in an obvious manner.

The entropy of moist air per unit weight of dry air is subject to augmentation by amount $\bar{s}_a + W\bar{s}_w$, where $\bar{s}_a$ and $\bar{s}_w$ are arbitrary constants. In Table 13 the constant $\bar{s}_a$ has been adjusted so as to make the specific entropy of dry air zero at 0°C and atmospheric pressure; the constant $\bar{s}_w$ has been adjusted so as to make the specific entropy of saturated liquid water zero at 0°C in conformity with the usual steam table practice.

Condensed Water

In the definition of thermodynamic wet bulb temperature to be given later there appears the specific enthalpy h'_w of pure (air-free) compressed liquid (or solid) water. Values of h'_w and of the corresponding specific entropy s'_w are listed in Table 14 for standard atmospheric pressure. The reference point at which both enthalpy and entropy are assigned the value zero is saturated liquid at 0°C in conformity with usual steam table practice. The values for undercooled liquid are based in part on an extrapolation of the Smith and Keyes[28] volume data, in part on the recommended extrapolation of the Goff-Gratch formula in Equation 6 for the saturation pressure p_{sf}, and in part on other data previously quoted. Since the first extrapolation is a fairly safe one inasmuch as the final results depend only slightly on the liquid volume, these values should be recommended for adoption, pending further research, if for no other reason than that they are consistent with previous recommendations.

Reference Points

The Working Subcommittee discussed the question of reference points at some length. Moist air, as defined in this report, is a mixture of five constituent gases: oxygen (O_2), nitrogen (N_2), argon (A), carbon dioxide (CO_2), and water vapor (H_2O). The enthalpy of each of these constituent gases i is subject to augmentation by a constant amount $\bar{h}_i$, called its null-point ($p=0, T=0$) enthalpy; the entropy of each constituent is subject to augmentation by a constant amount $\bar{s}_i$, called its null-point reduced entropy (by reduced entropy is meant the sum, $s+R\ln p$, with pressure expressed in standard atmospheres). It follows that the enthalpy and entropy of dry air are subject to augmentation by constant amounts, $\bar{h}_a = \Sigma x_i \bar{h}_i$ and $\bar{s}_a = \Sigma x_i \bar{s}_i$, respectively; similarly, the enthalpy and entropy of water are subject to augmentation by constant amounts, $\bar{h}_w$ and $\bar{s}_w$, respectively.

Tables 1 and 2 of this report are based on the arbitrary assignments: $\bar{h}_a = \bar{s}_a = \bar{h}_w = \bar{s}_w = 0$.

Tables 12, 13, 14 are based on the following assignments: for dry air, both enthalpy and entropy are assigned the value zero at 0°C, atmospheric pressure; for water, both enthalpy and entropy are assigned the value zero at 0°C, saturated liquid. Using the data previously quoted in this report it is easy to compute that these assignments are to be implemented by making: $\bar{h}_a = -65.260$ kcal/kg, $\bar{s}_a = -1.61712$ kcal/kg°K, $\bar{h}_w = +477.277$ kcal/kg, $\bar{s}_w = -0.83960$ kcal/kg°K.

The Goff-Gratch tables, Thermodynamic Properties of Moist Air[16] and Thermodynamic Properties of Water at Saturation[17], are based on the following assignments: for dry air, both enthalpy and entropy are assigned the value zero at 0°F, atmospheric pressure; for water, both enthalpy and entropy are assigned the value zero at 32 F, saturated liquid. Using the data previously quoted in this report it is easy to compute that these assignments are to be implemented by making: $\bar{h}_a = -109.782$ Btu/lb, $\bar{s}_a = -1.60096$ Btu/lb°R, $\bar{h}_w = +859.099$ Btu/lb, $\bar{s}_w = -0.83960$ Btu/lb°R. The tables referred to have been reproduced in HEATING, VENTILATING, AIR CONDITIONING GUIDE published and widely distributed by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS. Using the same assignments, additional tables giving the thermodynamic properties of moist air for six different pressures above

Table 11—Volume of Moist Air per Unit Weight of Dry Air *(Standard Atmospheric Pressure)* **(cm³/g)**

t °C	v_a	Tol. ±	v_{af}	Tol. + −	v_{ai}	Tol. + −
-90	517.13	25	-	-	517.13	25
-80	545.69	23	-	-	545.69	23
-70	574.20	22	-	-	574.20	22
-60	602.70	21	-	-	602.71	21
-50	631.18	20	631.22	-	631.20	20
-40	659.65	19	659.77	-	659.73	19
-30	688.09	18	688.44	-	688.35	18
-20	716.53	18	717.42	-	717.26	18
-10	744.96	17	747.06	-	746.88	18
0	773.36	17	778.06	18	778.06	18
10	801.77	17	811.61	19	-	-
20	830.18	17	849.80	21	-	-
30	858.57	18	896.13	25	-	-
40	886.96	18	956.67	29	-	-
50	915.35	19	1042.38	38	-	-
60	943.73	19	1174.90	56	-	-
70	972.10	20	1404.25	93	-	-
80	1000.48	20	1879.2	19	-	-
90	1028.85	21	3341.2	64	-	-

Table 12—Enthalpy of Moist Air per Unit Weight of Dry Air *(Standard Atmospheric Pressure)* **(kcal/kg)**

t °C	h_a	Tol. ±	h_{af}	Tol. + −	h_{ai}	Tol. + −
-90	-21.641	20	-	-	-21.641	20
-80	-19.230	17	-	-	-19.230	17
-70	-16.823	14	-	-	-16.822	14
-60	-14.418	11	-	-	-14.414	11
-50	-12.0132	89	-11.9897	-	-11.9992	90
-40	- 9.6093	68	- 9.5410	-	- 9.5634	71
-30	- 7.2065	49	- 7.0226	-	- 7.0696	55
-20	- 4.8046	31	- 4.3487	-	- 4.4298	46
-10	- 2.4027	16	- 1.3540	-	- 1.4510	45
0	0.0000	0	+ 2.2622	45	+ 2.2622	45
10	2.4021	15	7.0091	105	-	-
20	4.8052	28	13.744	20	-	-
30	7.2096	42	23.880	34	-	-
40	9.6140	55	39.796	54	-	-
50	12.0195	67	65.731	87	-	-
60	14.4264	80	110.00	16	-	-
70	16.8348	91	191.71	31	-	-
80	19.2447	104	367.68	67	-	-
90	21.657	12	921.1	24	-	-

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Thermodynamic Properties of Moist Air

Standard Atmospheric Pressure

t(°C)	h_a	h_s	s_a	s_s	h'_w	s'_w
—90	43.619	43.619	1.52101	1.52101	—595.07	0.3806
0	65.260	65.714	1.61712	1.62030	—477.26	0.8396
90	86.917	310.54	1.68564	2.87451	—387.23	1.1244

atmospheric and up to 150 lb/in.[2] have been compiled in the University of Pennsylvania Thermodynamics Research Laboratory[20] under Project G-8. It is clear, therefore, that the assignments of this paragraph are destined to be used rather widely in engineering circles.

On the other hand, the tables being compiled in the University of Pennsylvania Thermodynamics Research Laboratory for publication in Smithsonian Meteorological Tables[21] are based on the following assignments: for dry air, both enthalpy and reduced entropy (by reduced entropy is *here* meant the sum, $s+R\ln p$, with pressure expressed in *millibars*) are assigned the value zero at 0°C, zero pressure; for water, both enthalpy and entropy are assigned the value zero at 0°C, saturated liquid. Using the data previously quoted in this report it is easy to compute that these assignments are to be implemented by making: $\overline{h}_a = -65.326$ kcal/kg, $\overline{s}_a = -2.09179$ kcal/kg°K, $\overline{h}_w = +477.277$ kcal/kg, $\overline{s}_w = -0.83960$ kcal/kg°K.

In most, if not all, tables giving the thermodynamic properties of moist air, the usual steam tables practice of assigning the value zero to both the enthalpy and the entropy of water at 0°C, saturated liquid is followed. There is, however, no unanimity in the choice of reference point for dry air. Perhaps this lack of unanimity is of little practical importance because it is so easy to convert from one choice to another. The Working Subcommittee decided not to indicate a preference.

Recent developments in the science of interpreting atomic and molecular spectra have focused attention upon the zero-pressure isobar as the line along which the thermodynamic properties of gases and gas mixtures generally can at any time be known with greatest accuracy. This is certainly true in the case of moist air; hence the relative permanence of the numerical values assigned to the null-point enthalpies and reduced entropies of dry air and water would be enhanced by placing both reference points on the zero-pressure isobar at some stated value(s) of absolute temperature. The obvious location would be the null-point ($p=0, T=0$) itself; indeed the null-point suggests itself as the logical reference point for all gases.

Even if the null-point were to be adopted as the common reference point it would still be necessary to decide what are the best values to assign to the enthalpy and reduced entropy there for each gas under consideration. Tables 1 and 2 of this report are based on the assignments, $\overline{h}_a = \overline{s}_a = \overline{h}_w = \overline{s}_w = 0$, as is the preceding skeleton table.

In this table the unit of enthalpy is the kcal/kg, that of entropy, the kcal/kg°K. At —90°C the values given for h_s and s_s are for saturation with respect to ice; those for h'_w and s'_w at this temperature are for ice also. The data in this table should be compared with those in Tables 12, 13, and 14.

According to the so-called Third Law of Thermodynamics the assignments, $\overline{s}_i = 0$, are always permissible if nuclear changes are ruled out of consideration provided that, in computing values of $s^\circ_i - \overline{s}_i$, possible contributions from nuclear spin and other nuclear energy levels are consistently included or excluded (it is common practice to exclude). But the assignments, $\overline{h}_i = 0$, are in general not permitted if ordinary chemical changes are not to be ruled out of consideration. In this case only as many of the $\overline{h}_i$ can be disposed of arbitrarily as there are distinct atomic species. It would therefore seem logical to exercise this permitted arbitrariness on the null-point enthalpies of the monatomic gases themselves. Undoubtedly there will be a move in this direction as the accuracy and completeness of our knowledge of the dissociation energies of molecular gases, by which their null-

Table 13—Entropy of Moist Air per Unit Weight of Dry Air *(Standard Atmospheric Pressure)* **(kcal/kg°K)**

t °C	s_a	Tol. ±	s_{af}	Tol. ±	s_{a1}	Tol. ±
-90	-0.09611	11	-	-	-0.09611	11
-80	-0.08332	9	-	-	-0.08332	9
-70	-0.07116	7	-	-	-0.07116	7
-60	-0.05960	5	-	-	-0.05958	5
-50	-0.04957	4	-0.04846	-	-0.04850	4
-40	-0.03804	3	-0.03774	-	-0.03783	3
-30	-0.02795	2	-0.02716	-	-0.02735	2
-20	-0.01826	1	-0.01638	-	-0.01670	2
-10	-0.00896	1	-0.00478	-	-0.00514	2
0	0.00000	0	+0.00870	2	+0.00870	2
10	+0.00864	1	0.02574	4	-	-
20	0.01698	1	0.04911	7	-	-
30	0.02504	2	0.08312	12	-	-
40	0.03284	2	0.13485	18	-	-
50	0.04040	2	0.21669	29	-	-
60	0.04774	3	0.35246	52	-	-
70	0.05486	3	0.59601	95	-	-
80	0.06178	4	1.11236	203	-	-
90	0.06852	4	2.67830	71	-	-

Table 14—Properties of Condensed Water *(Standard Atmospheric Pressure)*

t °C	Enthalpy, h'_w (kcal/kg) liquid	Enthalpy, h'_w (kcal/kg) solid	Entropy, s'_w (kcal/kg°K) liquid	Entropy, s'_w (kcal/kg°K) solid
-90	-	-117.79	-	-0.4590
-80	-	-114.25	-	-0.4402
-70	-	-110.54	-	-0.4214
-60	-	-106.64	-	-0.4028
-50	-53.97	-102.58	-0.2192	-0.3841
-40	-42.00	- 98.34	-0.1666	-0.3655
-30	-30.96	- 93.92	-0.1201	-0.3470
-20	-20.40	- 89.34	-0.0776	-0.3285
-10	-10.14	- 84.57	-0.0378	-0.3101
0	+ 0.02	- 79.64	0.0000	-0.2916
10	10.06	-	+0.0361	-
20	20.06	-	0.0708	-
30	30.04	-	0.1043	-
40	40.03	-	0.1367	-
50	50.01	-	0.1681	-
60	60.00	-	0.1985	-
70	69.99	-	0.2281	-
80	80.01	-	0.2568	-
90	90.05	-	0.2848	-

point enthalpies are related to those of the associated monatomic gases, become more satisfactory.

Dew-Point Temperature

A derived thermodynamic property of moist air of great importance in hygrometry is the socalled thermodynamic dew point temperature T_d. The Working Subcommittee recommends that this important property be defined as the solution $T_d(p,W)$ of the equation,

$$W_s(p,T_d) = W \quad \ldots\ldots\ldots\ldots\ldots\ldots (9)$$

The definition is given in mathematical form to avoid ambiguity. It is a problem belonging properly to practical hygrometry to design and construct dew point hygrometers capable of reading values of thermodynamic dew point temperature as defined by Equation 9 after suitable calibration. The IMO has concurred in the foregoing definition.

Wet Bulb Temperature

Another derived thermodynamic property of moist air of great importance in psychrometry is the so-called thermodynamic wet bulb temperature T_w. The Working Subcommittee recommends that this important property be defined as the solution $T_w(p,T,W)$ of

$$h(p,T,W) - W \cdot h'_w(p,T_w) = h_s(p,T_w) - W_s(p,T_w) \cdot h'_w(p,T_w) \ldots\ldots (10)$$

where it is to be understood that h'_w denotes the specific enthalpy of pure compressed liquid (or solid) water. This definition is also given in mathematical form to avoid ambiguity. It is a problem belonging properly to practical psychrometry to design and construct psychrometers capable of reading values of thermodynamic wet bulb temperature as defined by Equation 10 after suitable calibration. It seems likely that the IMO will concur in the foregoing definition.

In justification of this definition it may be said that it combines the correct energy and weight accountings for the steady-flow process of injecting pure compressed liquid (or solid) water at pressure p and temperature T_w into a stream of moist air at pressure p, temperature T, and humidity ratio W to bring the air adiabatically to saturation at pressure p and temperature T_w. This is to be regarded as the appropriate idealization of the actual process by which the thin film of water on a wet bulb thermometer immersed in a stream of moist air maintains, at any rate for a time, a steady value below that of the air itself.

Acknowledgment

It is a pleasure to acknowledge the substantial cooperation of the Navy Department, Bureau of Ships, in the preparation of this report. It was under the sponsorship of the Bureau that the University of Pennsylvania Thermodynamics Research Laboratory was operated from July 1, 1945 until June 30, 1948, when cognizance was transferred to the Office of Naval Research. The invaluable aid of S. Gratch, assistant professor of Mechanical Engineering, Towne Scientific School, University of Pennsylvania, and Project Leader in charge of gas properties research in the University of Pennsylvania Thermodynamics Research Laboratory is also gratefully acknowledged.

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20. On the Specific Heat of Supercooled Water, by H. T. Barnes and H. L. Cooke. (*Physical Reviews*, Vol. 15, 1902, pp.65-72.)

21. Heat of Fusion of Ice. A Revision, by N. S. Osborne. (*Journal Research National Bureau of Standards*, RP1260, Vol. 23, 1939, pp.643-646.)

22. The Entropy of Water and the Third Law of Thermodynamics. The Heat Capacity of Ice from 15 to 273°K, by W. F. Giauque and J. W. Stout. (*Journal American Chemical Society*, Vol. 58, 1936, p.1144.)

23. Vapor Pressure of Subcooled Liquid Water to —50°C, by S. Gratch. (Special Report, University of Pennsylvania Thermodynamics Research Lab., August 1947.)

24. Measurements of Heat Capacity and Heat of Vaporization of Water in the Range 0° to 100°C, by N. S. Osborne, H. F. Stimson, and D. C. Ginnings. (*Journal Research National Bureau of Standards*, RP1228, Vol. 23, 1939, pp.197-260.)

25. Bestimmung des Sättigungsdruckes von Wasserdampf unter 0°, by K. Scheel and W. Heuse. (*Annalen der Physik*, Vol. 29, 1909, pp.723-737.)

26. The Volumes of Unit Mass of Liquid Water and their Correlation as a Function of Pressure and Temperature, by L. B. Smith and F. G. Keyes. (*Proceedings American Academy of Arts and Sciences*, Vol. 69, 1934, pp.285-314.)

27. The Thermodynamic Properties of Water Substance 0° to 150°C, by F. G. Keyes. (*Journal chem. Phys.*, Vol. 15, 1947, p.602.)

28. The Location of the Normal Sulfur and Mercury Boiling Points on the Thermodynamic Temperature Scale, by B. E. Blaisdell and J. Kaye. (*Temperature, Its Measurement and Control in Science and Industry*, Reinhold Publishing Co., New York, 1941, pp.127-140.)

29. The Third International Conference on Steam Tables. (*Mechanical Engineering*, Vol. 57, 1935, pp.710-713.)

30. The Humidity Ratio (Mixing Ratio) of Moist Air at Saturation, by J. A. Goff and S. Gratch. (Special Report, University of Pennsylvania Thermodynamics Research Laboratory, March 1948.)

31. Compilation of Properties of Moist Air for the Smithsonian Meteorological Tables, by S. Gratch. (Progress Report, University of Pennsylvania Thermodynamics Research Laboratory, August 1948.)

32. *International Critical Tables*, Vol. 3, 1928, p.257.

33. Thermal Properties of Saturated Water and Steam, by N. S. Osborne, H. F. Stimson, and D. C. Ginnings. (*Journal Research National Bureau of Standards*, RP1229, Vol. 23, 1939, pp.261-270.)

34. Wärmedehnung des Eises Zwischen 0 und —253°, by M. Jakob and S. Erk. (*Wissenschaftliche Abhandlungen der Physikalisch Technischen Reichsanstalt*, Vol. 12, 1929, pp.301-316.)

35. *Thermodynamic Properties of Steam*, by J. H. Keenan and F. G. Keyes. (John Wiley and Sons, Inc., New York, 1936.)

36. Goff Chart for Moist Air, by S. Gratch. (Summary Report of Project G-8, University of Pennsylvania Thermodynamics Research Laboratory, April 1946, 1947, 1948.)

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Honored by Life Membership

Nate W. Downes
Sacramento, Calif.

NATE W. DOWNES

— Supervising engineer, maintenance and operation, Dept. of Public Works, State of California, Sacramento.

— Kansas City, Missouri School District for 31 years as assistant chief engineer, chief engineer, superintendent of buildings, assistant superintendent of schools.

— Engineer Corps, U.S. Army, 1918.

— Consulting engineer, 1919 to 1922.

ROBERT P. SCHOENIJAHN

— Consulting engineer, Wilmington, Del. for 32 years. Specializes in mechanical, electrical and sanitary engineering.

— Two years as chief engineer for mechanical and electrical equipment for city and county buildings, Wilmington, New Castle County.

— Experience in U.S. explosive plant, 1918-19; W. S. Timmis, N. Y., designer and draftsman; Marks and Woodwell, N. Y., designing engineer.

— Educated in Brooklyn public schools; Cornell University, Ithaca, N. Y. Member *ASME*.

Robert P. Schoenijahn

Wilmington

Delaware

RALPH S. FRANKLIN

Ralph S. Franklin

Boston

Massachusetts

— Mechanical engineer, Lockwood Greene Engineers, Inc., Boston, Mass.

— Formerly president, A. B. Franklin, Inc., Boston; specialty: heating, ventilating, and industrial drying.

— Designed heating and ventilating plants for Soldiers and Sailors Memorial Building, Melrose, Mass.; Lakeville Sanitarium; So. Norwalk High School Building, Conn.; Gedik Pasha School, Constantinople, Turkey.

— Educated at Massachusetts Institute of Technology, Cambridge.

EDWARD B. JOHNSON

— Designer, mechanical engineer, Parsons, Brinckerhoff, Hall & MacDonald, New York City.

— Thirty-five years' experience with steam and hot water heating of homes and buildings.

— Designed heating plants for U.S. Army Engineers Building at Harmon Field, and Keflavik, Iceland Airfields.

— Secretary of New York Chapter ASHVE, 1922-30 and 1931-32.

— Other experience: In 1903-05, with Rodman Rapid Transit Commission, constructing first subway in New York City; engineer, American Radiator Co., New York, N. Y.; 4 years, Bethlehem Steel Co., Mariners Harbor, S. I., N. Y.

E. B. Johnson
New York, N. Y.

JOURNAL SECTION

Honored by Life Membership

ARTHUR L. SANFORD

— Mechanical engineer, C. H. Johnston, architects and engineers, St. Paul, Minn.

— More than 40 years of design, construction, installation and inspection of mechanical equipment of buildings, including heating, ventilating, air conditioning, electrical work and power plants.

— Past president Minnesota Chapter ASHVE; registered professional engineer, State of Minnesota.

— Other experience: Minneapolis Board of Education; Minneapolis-Honeywell Regulator Co., Minneapolis; B. F. Sturtevant Co., Boston and Chicago; B. G. Carpenter Co., Wilkes Barre, Pa.

Arthur L. Sanford
St. Paul, Minn.

Eugene P. Bradley
St. Louis, Mo.

EUGENE P. BRADLEY

— President, Hester-Bradley Co., St. Louis, Mo., for many years.

— Helped establish St. Louis Chapter, ASHVE; served as the Chapter's second president.

— Presented a paper, *Mutual Responsibility of the Heating Engineer and the Architect* at the Semi-Annual Meeting, Buffalo and Detroit, June, 1922.

— Was on ASHVE Committees, 1921-24: Legislation, Nominating, Chapters, Standards of Ventilation.

— Served in U.S. Army as Captain, Construction Division. Chief mechanical engineer for firms engaged in construction and installations in air fields, camps and ordnance plants.

EDWARD E. ASHLEY

— Consulting engineer, in New York City.

— Formerly sales engineer, Mercury Manufacturing Co., Chicago; consulting, mechanical and electrical engineer, Starrett and Van Vleck, architects, New York City.

— During World War I served as Captain in the Air Service of the U.S. Army.

— ASHVE representative on *ASA* Sectional Committee on Graphical Symbols and Abbreviations.

Edward E. Ashley

New York

New York

Walter W. Nobbs
London, England

WALTER W. NOBBS

— Consulting engineer in heating, ventilating and water supplies, London, England. Formerly civil engineer to British Air Ministry.

— Earlier experience with Purcell & Nobbs successively as fitter, foreman, draftsman, technical advisor and manager.

— Past President, Silver Medallist, *IHVE;* member of *Royal Society of Arts, Institute of Mechanical Engineers.*

— Consulting engineer in construction of The County Hall, London; International Municipal Buildings, Shanghai.

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JOURNAL SECTION

Honored by Life Membership

William Forsberg
New London, Conn.

WILLIAM FORSBERG

— Secretary, The Hopson & Chapin Manufacturing Co., New London, Conn.
— Formerly, draftsman, salesman and superintendent, The Hopson & Chapin Manufacturing Co., design and sales of heating apparatus, and supervision of installations.
— Specialist in the heating of country houses and in central station heating plants.
— Born in New London, Conn., 1884; member of Connecticut Chapter.

WILLIAM MALLIS

— Heads William Mallis Co., Seattle, Wash. Specializes in heating and ventilating of school buildings.
— Apprentice in architecture in Scotland; in 1909 became associated with J. H. Felt Co., Kansas City, Mo.
— Nearly 50 years of engineering and architecture; designed heating and ventilating plants for many high schools in Missouri.
— Born in Scotland; educated in Morrison's Academy, Scotland.

William Mallis
Seattle
Washington

HENRY P. REGER

Henry P. Reger
Chicago
Illinois

— President, H. P. Reger Co., Chicago, Ill.
— Contractor for heating of Merchandise Mart, YMCA Building, U.S. Post Office, Illinois Soldiers and Sailors Home, etc.
— Member of *Heating, Piping and Air Conditioning Contractors National Association,; National Association of Master Plumbers.*
— Born in Mineral Point, Wis., 1876.

CHARLES B. KORN

— Partner, Reber-Korn Co., Allentown, Pa.
— Forty-five years in the heating and ventilating field; heating, high pressure and power piping, sprinkling systems.
— Formerly with Hersh Bros. Co., Allentown, as superintendent and engineer; William H. Taylor & Co. as draftsman and assistant engineer.
— Educated in Keystone State Normal School; mechanical engineering course, Scranton, Pa.

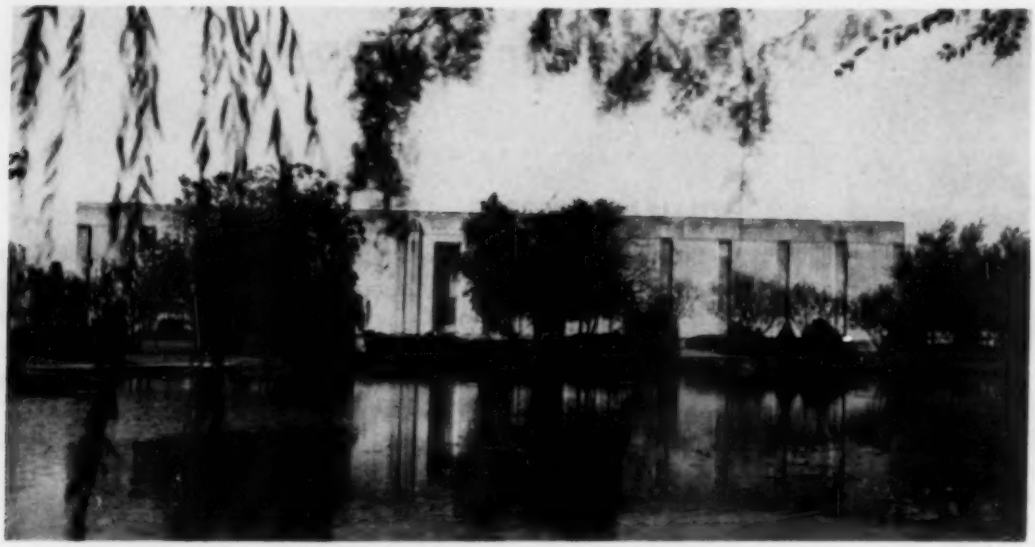
Charles B. Korn
Allentown
Pennsylvania

H. M. Nobis
Cleveland, Ohio

H. M. NOBIS

— Plant engineer, Warner & Swasey Co., Cleveland, Ohio.
— More than 50 years of experience as consulting engineer, contractor, designer. Holds several patents in heating and ventilating field.
— Was engineer in Europe and Canada before setting up consulting engineer's office in 1911.
— Naval experience: Steam plant equipment of cruisers.
— Lives in East Cleveland, Ohio, member of Northern Ohio Chapter.

JOURNAL SECTION



Dallas Museum of Natural History

Technical Topics to Compete With Typical Texas Show at 56th Annual Meeting

At the 56th Annual Meeting of ASHVE to be held in Dallas, Tex., January 22-27, 1950, five technical sessions will extend over a four day period, with each session built around a particular engineering topic, C. R. Gardner, general chairman of the Committee on Arrangements, announced.

The five engineering topics are: (1) Heat and Moisture Transfer; (2) Heat Pump; (3) Panel Heating; (4) Air Distribution; (5) Research Laboratory and Cooperative Research Session. There will be at least three technical papers per session, presented by well known engineers, professors, and scientists.

A Bear By the Tail

The North Texas Chapter, in making ready for the 56th Annual Meeting with its multitude of arrangements and accommodations has "a bear by the tail," said G. A. Linskie, president of the Chapter. A social program is being planned, Mr. Gardner stated, which gives promise of being a typical Texas show—that is: "The biggest, the best, most outstanding show ever presented." Both Mr. Linskie and Mr. Gardner praised the efforts of the tireless committeemen who are working hard to arrange the details of the 1950 Annual Meeting.

Train Schedules Announced

Since Daylight Saving Time ended in September, changes have been made in train schedules. Members are urged to check their local railroad agents for any changes in schedules which may occur in the future. The schedules which appear on page 133 are in effect at the present time.

Restored cabin of the founder of Dallas

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Train Schedules via St. Louis

Lv.	New York	B. & O.	3:05 P.M. Jan. 20
Lv.	Philadelphia		5:38 P M. Jan. 20
Lv.	Baltimore		7:25 P.M. Jan. 20
Lv.	Washington		8:30 P.M. Jan. 20
Lv.	Cincinnati		9:10 A.M. Jan. 21
Ar.	St. Louis		3:15 P.M. Jan. 21
Lv.	New York	N. Y.C.	7:15 P.M. Jan. 20
Lv.	Boston		4:40 P.M. Jan. 20
Lv.	Syracuse		12:30 A.M. Jan. 21
Lv.	Rochester		1:08 A.M. Jan. 21
Ar.	Buffalo		2:20 A.M. Jan. 21
Lv.	Buffalo		2:57 A.M. Jan. 21
Lv.	Cleveland		6:10 A.M. Jan. 21
Lv.	Indianapolis		10:50 A.M. Jan. 21
Ar.	St. Louis		3:50 P.M. Jan. 21
Lv.	New York	Penna.	7:35 P.M. Jan. 20
Lv.	Philadelphia		9:06 P.M. Jan. 20
Lv.	Pittsburgh		4:00 A.M. Jan. 21
Lv.	Columbus		8:00 A.M. Jan. 21
Lv.	Indianapolis		10:32 A.M. Jan. 21
Ar.	St. Louis		3:00 P.M. Jan. 21
Lv.	Norfolk	C. & O.	2:20 P.M. Jan. 20
Ar.	St. Louis	N. Y. C.	3:50 P.M. Jan. 21
Lv.	Detroit	N. Y. C.	10:25 P.M. Jan. 20
Ar.	St. Louis		11:30 A.M. Jan. 21
Lv.	Grand Rapids	C. & O.	7:20 A.M. Jan. 21
Ar.	Chicago		10:20 A.M. Jan. 21
Lv.	Montreal	Can. Pac.	3:30 P M. Jan. 20
Lv.	Toronto	Can. Pac.	10:00 P.M. Jan. 20
Ar.	Chicago	N. Y. C.	8:10 A.M. Jan. 21
Lv.	Milwaukee	C.M.St.P. & P.	9:15 A.M. Jan. 21
Ar.	Chicago		10:35 A.M. Jan. 21
Lv.	Chicago	G.M. & O.	11:45 A.M. Jan. 21
Ar.	St. Louis		4:55 P.M. Jan. 21

ASHVE Special

Lv.	ST. LOUIS	FRISCO-M.K.T.	5:30 P.M. JAN. 21
Ar.	DALLAS	FRISCO-M.K.T.	7:35 A.M. JAN. 22

Southeastern Points Direct to Dallas

Lv.	Durham	So. Ry.	7:15 P.M. Jan. 20
Ar.	Greensboro		9:05 P.M. Jan. 20
Lv.	Greensboro	So. Ry.	9:20 P.M. Jan. 20
Ar.	Atlanta		6:30 A.M. Jan. 21
Lv.	Atlanta	So. Ry.	7:30 A.M. Jan. 21
Ar.	Birmingham		11:30 A.M. Jan. 21
Lv.	Birmingham	Frisco	11:50 A.M. Jan. 21
Ar.	Memphis		6:30 P.M. Jan. 21
Lv.	Memphis	Mo. Pac.	8:30 P.M. Jan. 21
Ar.	Dallas		7:30 A.M. Jan. 22
Lv.	New Orleans	T. & P.	8:05 P.M. Jan. 21
Lv.	Shreveport		5:00 A.M. Jan. 22
Ar.	Dallas		9:30 A.M. Jan. 22

Southwestern Points Direct to Dallas

Lv.	Tulsa	Frisco	10:30 P.M. Jan. 21
Ar.	Dallas		7:45 A.M. Jan. 22
Lv.	Oklahoma City	Santa Fe	12:35 A.M. Jan. 22
Ar.	Dallas		8:00 A.M. Jan. 22
Lv.	Houston	So. Pac.	11:30 P.M. Jan. 21
Ar.	Dallas		6:55 A.M. Jan. 22
Lv.	San Antonio	M.K.T.	11:00 P.M. Jan. 21
Ar.	Dallas		7:15 A.M. Jan. 22

Western Points Direct to Dallas

Lv.	San Francisco	So. Pac.	9:00 P.M. Jan. 19
Lv.	Los Angeles	So. Pac.	3:00 P.M. Jan. 20
Ar.	Dallas	T. & P.	7:05 A.M. Jan. 22
Lv.	Seattle	Un. Pac.	4:30 P.M. Jan. 19
Lv.	Portland		10:00 P.M. Jan. 19
Ar.	Denver		8:10 A.M. Jan. 21
Lv.	San Francisco	So. Pac.	6:00 P.M. Jan. 19
Lv.	Salt Lake City	Un. Pac.	5:45 P.M. Jan. 20
Ar.	Denver		7:45 A.M. Jan. 21
Lv.	Denver	C. & S.-F.W.D.C.	12:30 P.M. Jan. 21
Ar.	Dallas		7:15 A.M. Jan. 22
Lv.	Winnipeg	Gr. Nor.	7:30 P.M. Jan. 20
Ar.	Minneapolis		7:25 A.M. Jan. 21
Lv.	Minneapolis	R.I.	12:01 P.M. Jan. 21
Lv.	Des Moines		5:03 P.M. Jan. 21
Ar.	Kansas City		9:15 P.M. Jan. 21
Lv.	Omaha	C.B. & Q.	4:15 P.M. Jan. 21
Ar.	Kansas City		9:25 P.M. Jan. 21
Lv.	Kansas City	M.K.T.	9:35 P.M. Jan. 21
Ar.	Dallas		8:40 A.M. Jan. 22

Consider Speakers for Banquet and Luncheon

Several prominent public figures are being considered for the coveted roles of guest speaker during the Wednesday night banquet and guest speaker for the welcome luncheon. Among the speakers being considered are several who have distinguished themselves in the fields of education, politics and industry.

During the week of the 56th Annual Meeting, Lester T. Avery, First Vice President, ASHVE, will make a guest appearance before nearly 500 members of the Dallas Rotary Club on January 25, 1950. Mr. Avery will speak on the subject, *Keeping Cool* and will also describe future trends in air conditioning.

Historical Interest

Hardly more than 100 years have passed since Dallas was a one room cabin on the Texas frontier. In that time it has grown to a city of expanding industry and a population of half a million.

Big "D"—the Southwest's affectionate name for Dallas—was born in 1841 when John Neely Bryan, a Tennesseean, came to Texas to set up a trading post. He was so impressed with the fertile spot on the banks of the Trinity River that he built for himself a shelter of cedar boughs. Around this single log cabin the settlement grew.

The cabin has been restored and is now standing on the grounds of the County Court House not far from its original site.

EXPOSITION SPACE RAPIDLY BEING FILLED

Nearly 200 manufacturers are included on the list of exhibitors at the Southwest Air Conditioning Exposition, which will be held simultaneously with the 56th Annual Meeting, ASHVE. Almost all of the available space is assigned in the Hall of Agriculture and the Poultry Building at Fair Park where the Exposition is being held.

The application of air conditioning to large industrial, business and institutional building projects will be featured at the Exposition. New designs in great variety, including summer cooling units, room coolers, fans, and controls will be exhibited. The Exposition is being managed by the International Exposition Co., New York, N. Y. Charles F. Roth is manager and E. K. Stevens associate manager. Both are members of ASHVE.

McKEEMAN LEAVES ASHVE POST FOR PROFESSORSHIP

C. A. McKeeman, since 1946 assistant to the president, ASHVE, has resigned to take a position at North Carolina State College, Raleigh, N. C. Mr. McKeeman will have charge of the new curriculum in heating and air conditioning which is being offered this year for the first time. The post which he holds is the L. L. Vaughan Professorship in mechanical engineering, named in honor of Prof. L. L. Vaughan, now retired from the State College faculty (See August 1949 JOURNAL, p. 136).

Clyde A. McKeeman
Raleigh, N. C.

The new teaching program which Mr. McKeeman directs will cover a four year period and will lead to the B.S. degree in heating and air conditioning. Short courses will also be offered for industrial and experimental work.

Mr. McKeeman, a member of ASHVE, was on the faculty of *Case Institute of Technology*, Cleveland, Ohio, from 1926 to 1944. He then became director of supervisory training for Thompson Products and Subsidiaries and remained there until 1946, when he became assistant to the president, ASHVE.

Summary of Chapter Meetings*

• **DELTA**—September 13, 1949. A local member of ASHVE was guest speaker at the September meeting held in New Orleans. Illustrating his talk with slides and charts, Walter Cooke spoke on the characteristics of various types of fans and blowers.

The subject of elections was discussed, and Pres. R. B. Guest appointed J. S. Burke to study the By-Laws and make a report. Another talk was given by H. L. Salaun on memberships. An announcement was made regarding the 56th Annual Meeting to be held in Dallas, January 23-27, 1950. A film of the 1948 World Series was shown by the Entertainment Committee. Attendance 47. Attendance ratio 0.43.

• **NEBRASKA**—June 17, 1949. The meeting was an outing and dinner dance for members, guests and their wives and was held at the Manawa Country Club, Council Bluffs, Iowa. The 1949-50 officers were elected as follows: *President*—C. A. Carter; *Vice President*—G. W. Colburn; *Secretary*—K. E. Martin; *Treasurer*—S. W. Black. Attendance 66.

• **NORTH TEXAS**—August 15, 1949. The problems of altitude and varying temperatures were discussed by three speakers at the August meeting in Dallas. P. N. Vinther spoke about a high velocity air conditioning system. He described the problem of high outside wet bulb temperatures and low altitudes. Mr. McFadden spoke about the design of a hospital which had to take into consideration high altitudes and low outside wet bulb temperatures. Herman Blum gave a talk on high velocity air conditioning systems using window units with fans. The speakers answered questions submitted from the floor.

C. R. Gardner introduced the speakers. Attendance 121. Attendance ratio 0.55.

• **NORTHERN OHIO** — June 10, 1949: Pres. Wesley R. Moore presided at a dinner meeting at the Manakiki Country Club, Cleveland, following an afternoon of golf in which 54 members and guests participated. The Entertainment Committee was headed by L. C. Burkes. A past president's certificate was presented to J. E. Wilhelm by President Moore. The officers for the coming year were installed and the gavel passed to R. L. Byers, incoming president. Attendance 72.

• **OKLAHOMA**—September 12, 1949. Members gathered in Oklahoma City for the September meeting were transported in imagination to Saudi Arabia by W. H. Carson, Dean of Engineering, University of Oklahoma, Norman. Dean Carson's talk about that country covered the central type of refrigeration using chilled water piped to the various buildings and houses. He described the use of airplanes to service exploration parties, and discussed the problems encountered in a large operation in the desert.

F. X. Loeffler, Jr. reported that the next meeting would be held October 5 and that Cyril Tasker of the ASHVE Research Laboratory would speak. Attendance 34.

• **ST. LOUIS**—September 13, 1949. Wind tunnels and the flight of supersonic planes were described at the September meeting by Carl J. Wenzinger of Sverdrup and Parcel,

*Note: The attendance ratios shown represent the membership attendance divided by the chapter membership. These ratios will be useful as a partial indication of interest shown by local chapter members in various types of subjects programmed by the various chapters and may be useful in deciding on subjects for chapter meetings.

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Inc., consulting engineers, St. Louis. Mr. Wenzinger is head aerodynamicist of the engineering company. He described the various types of wind tunnels, opened and closed, and the subsonic, transonic, supersonic and hypersonic ranges of flight. Mach numbers and Reynolds numbers were prominent in the data. The speaker was introduced by J. H. Carter.

Guests at the meeting included H. D. Cook of the Wisconsin Chapter of ASHVE, and Hendrik Manden of Enkoping, Sweden. A communication was read concerning the 56th Annual Meeting to be held in Dallas next January. It was suggested that the members travel by rail, using an "All St. Louis" car. Attendance 60. Attendance ratio 0.29.

• SHREVEPORT—September 15, 1949. Guest speaker at the September meeting was E. J. Stern of the Armstrong Cork Co., Lancaster, Pa. He spoke on the subject of insulation and its many uses and value to industry. Pres. A. H. Otto introduced the speaker and also reported on the Semi-Annual Meeting of ASHVE held in Minneapolis last June.

An announcement was made concerning the nomination of candidates for Chapter offices for the year 1950. It was also announced that the speaker for the October meeting would be Cyril Tasker, Director of Research, ASHVE Research Laboratory, Cleveland, Ohio, and that the meeting would be held in the Washington-Youree Hotel. Attendance 24.

• SOUTH TEXAS—August 26, 1949. Several speakers appeared on the program of the August meeting which was held in Houston. P. N. Vinther spoke on methods used to correct operating difficulties. Herman Blum spoke on air conditioning a multi-story building, discussing the treatment of interior zones. G. A. Linskie, president of the North Texas Chapter, described the problems encountered in arranging the 56th Annual Meeting in Dallas.

C. R. Gardner, member of ASHVE Council and General Chairman of the 56th Annual Meeting to be held next January, spoke on plans for the January Meeting and Exposition, and reported on the progress made by the Committee on Arrangements. R. F. Taylor acted as master of ceremonies and introduced the speakers of the evening. Attendance 95. Attendance ratio 0.34.

• SOUTHERN CALIFORNIA—September 21, 1949. *Automatic Temperature Control and Basic Controls System Design* were subjects presented by two speakers at the meeting in Beverly Hills. The speakers were: W. O. Stewart, manager, Johnson Service Co., Los Angeles; and F. R. Fitzgerald, sales engineer, Minneapolis - Honeywell Regulator Co., Minneapolis. Mr. Stewart presented the features of pneumatic controls and Mr. Fitzgerald handled the discussion on electric controls. Slides were used for illustration and a question and answer period followed the formal program.

• SOUTHWEST TEXAS — September 9, 1949. The use of refrigeration in a chemical plant was described by R. E. Geatreaus and H. J. Metz at the September meeting in Corpus Christi. Mr. Geatreaus is assistant plant engineer and Mr. Metz investment engineer, Celanese Corp., Corpus Christi. The production of aldehydes, alcohols and solvents was described and it was pointed out that by the use of refrigeration, operations could proceed at lower pressures and lower cost.

B. P. Fisher reported on the 56th Annual Meeting to be held in Dallas next January. R. F. Taylor told members about the coming Southwest Air Conditioning Exposition. Pres. G. R. Rhine reported on the program for an all-day meeting to be held in San Antonio, October 11. Attendance 32.

John R. Allen Memorial Library

The ASHVE Committee on Research gratefully acknowledges the following gifts to the John R. Allen Memorial Library, which is maintained at the Society's Research Laboratory, 7218 Euclid Ave., Cleveland 3, Ohio.

The Heat Pump, Its Practical Application, by J. B. Pinkerton. Gift of the author.

Proceedings—War Emergency Issues, 25-36, 1947—The Institution of Mechanical Engineers, Vol. 157. Gift of the Institution.

The presentation of textbooks and pamphlets in the field of heating, ventilating, and air conditioning and allied fields, by authors and others, will be welcomed by this library. Suitable book plates carry the name of the donor and date of the gift.

ASHVE APPOINTMENTS MADE

Members of ASHVE who have been recently appointed to serve on Society committees are: I. W. Cotton, Indianapolis, Ind., Publication Committee; on Technical Advisory Committees: Air Distribution—C. H. Randolph, Milwaukee, Wis.; Heat Pump—E. N. Kemler, Houston, Tex., and W. E. Zieber, York, Pa., (Committee on Research representative); Physiological Research—Craig L. Taylor, Los Angeles, Calif.; Air Cleaning—R. B. Crepps, Newark, Ohio; Sorbents—E. W. Gifford, Milwaukee, Wis. Amos Turk, New York, N. Y., will serve on the TAC on Odors.

Candidates for Membership

The Constitution of the Society, as now amended, requires the following mode of procedure in voting on applicants for membership in the Society. All applications for membership are to be sent to the Secretary and the names of applicants and their references shall be printed in the next issue of the JOURNAL of the Society or sent to the members in other approved manner as ordered by the Council. When the replies are received from references, the Candidate's application shall be submitted to and acted upon by the Committee on Admission and Advancement as soon as possible.

When the Committee on Admission and Advancement has acted favorably upon a Candidate's application and assigned his grade, the Council shall vote upon the election of the proposed Candidate for membership by letter ballot. During the past month 54 applications for membership have been received and the names of these men and their sponsors are published in the following list.

Members are requested to scrutinize the list with care. The Committee on Admission and Advancement, and in turn, the Council, urge members to assume their share of responsibility of receiving these candidates into membership by advising the Secretary promptly of any whose eligibility for membership is in any way questioned.

All correspondence in regard to such matters is strictly confidential, and is solely for the good of the Society, which it is the duty of every member to promote.

Unless objection is made by some member by November 21, 1949, these candidates will be balloted upon by the Council. Those elected to membership will be notified by the Secretary immediately after election.

ARNOLD, R. B., Sales Engr., N. O. Nelson Co., Houston, Tex. PROPOSERS: W. J. Way, E. J. Stansbury. SECONDERS: Norman Way, R. H. Crowe.

BAER, M. A., Testing, Johnson Service Co., Milwaukee, Wis. PROPOSERS: J. R. Vernon, M. M. Herrick. SECONDERS: H. W. Alyea, H. W. Schreiber.

BENOIT, W. J., Estimator, Construction & Sales, Gallaher & Speck, Chicago, Ill. PROPOSERS: C. W. DeLand, H. M. Hart. SECONDERS: C. W. Johnson, G. W. Bornquist.

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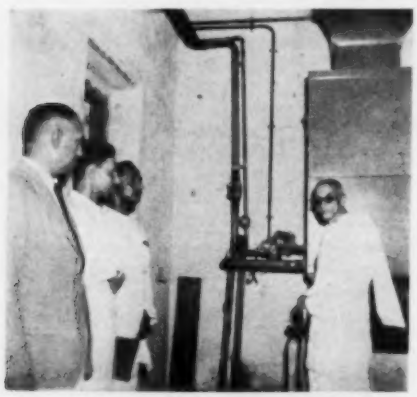
K. O. Werwath, president of the Milwaukee School of Engineering, Milwaukee, Wis., has been elected vice chairman of the Technical Institute Division of the *American Society for Engineering Education*.

E. C. Youngman has become a full partner in the firm of Mead & Hunt, consulting engineers, Madison, Wis. He was formerly mechanical engineer with the firm.

MEMBER FROM INDIA DESIGNS AIR CONDITIONING PLANTS

His Excellency C. Rajagopalachari, Governor General of India, is seen here opening one of the



multi-purpose cold storage plants installed for preservation of food and agricultural produce. On the center left is B. B. Ghosh, designing engineer. Mr. Ghosh, a member of the ASHVE, is responsible for the design of a number of cold storage and air conditioning plants in India. He visited the U.S. last year and attended the 55th Annual Meeting.

FORUM LAUNCHES CLIMATE CONTROL PROJECT

Several well known scientists appeared as speakers on the Climate Control Forum held recently at the Waldorf Astoria, New York City, under the sponsorship of the magazine, *House Beautiful*.

The physiological factors in design and construction for scientific climate control were described by Dr. L. P. Herrington, director of research, John B. Pierce Laboratory of Hygiene, New Haven, Conn. Dr. Paul Siple, military geographer, The General Staff, U. S. Army, Washington, D. C., spoke on climatological analysis of selected population centers of the continental United States in relation to design and construction factors.

Other speakers included industrial, research and architectural representatives. The group inspected a model home at Orange, N. J., to see how some of these principles were applied.

OBITUARIES

JOHN BAUER

John Bauer, Plattsmouth, Nebr., died recently at the age of 71.

He was born in 1878 at Plattsmouth and received his education in that city's public schools. He had about 50 years of experience in the heating and plumbing fields, designing and supervising the installation of heating plants of every type. At the time of his death he was manager of the Bauer Heating and Plumbing Co., Plattsmouth, Nebr.

EDWARD J. CALNAN

Edward J. Calnan, St. Catherines, Ont., Canada, was killed in a plane crash in Quebec on September 9, 1949. He was 48 years old.

Mr. Calnan had been superintendent of steam plants for the Ontario Paper Co., Ltd., Thorold, Ont., with supervision over the Thorold plant and a plant in Bay Combae, Que.

He was born at Grafton, Ont. and was graduated from the University of Toronto in 1928, with a degree of B.S. in mechanical engineering. Following his graduation, he entered the Bailey Meter Co., Cleveland, Ohio, as an engineer and was later transferred to the Bailey Meter Co., Ltd., of Montreal, Que. In 1930 he joined the staff of the Ontario Paper Co. at Thorold, Ont. and was superintendent of steam plants at the time of his death.

Mr. Calnan was active in sports and community life. While at the University of Toronto, he was on the football team and was also intercollegiate wrestling champion. At St. Catherines he organized and coached football for the Ontario Rugby and Football Union. For four years he was chairman of the Separate School Board of St. Catherines.

He is survived by his widow, a son and a daughter.

SHERMAN W. STROUSE

Sherman W. Strouse, Buffalo, N. Y., died on August 12, 1949 after having been ill for some time.

Mr. Strouse was manager of the Buffalo Sales Office of The Trane

S. W. Strouse
Buffalo, N. Y.

Co., La Crosse, Wis. He was born in 1900 at Albion, N. Y. and attended Buffalo Technical High School and Queens University. He received his B.S. degree in 1922.

He had many years of experience in the design, sale and supervision of installations of heating, ventilating, refrigeration and air conditioning equipment. He taught air conditioning and refrigeration courses at Buffalo Technical High School. He became affiliated with The Trane Co. in 1939.

He is survived by his widow and one daughter.

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Group B—Heat Transfer Between the Panel and the Space: J. W. James*, *Chairman;* H. F. Randolph, *Vice Chairman;* C. M. Ashley, J. T. Bergen, C. F. Boester, Wharton Clay, Linn Helander, A. L. Jaros, H. A. Lockhart, W. E. Long, C. O. Mackey, R. A. Miller, G. W. Penney, G. G. Sward, J. M. van Nieukerken, H. E. Wheeler.

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• **BALTIMORE:** Organized 1949. Headquarters, Baltimore, Md. Meets, Third Wednesday. PRESIDENT, E. L. Crosby, 2315 St. Paul St., Baltimore 18. SECRETARY, W. P. Flanigan, 5508 Greenleaf Rd., Baltimore 10.

• **CENTRAL NEW YORK:** Organized 1944. Headquarters, Syracuse, N. Y. President, V. S. Day, 300 S. Geddes St., Syracuse 1. Secretary, J. H. Carpenter, 3515 Midland Ave., Syracuse 7.

• **CENTRAL OHIO:** Organized 1944. Headquarters, Columbus, Ohio. Meets, Third Monday. PRESIDENT, E. A. Norman, Jr., 1150 Chesapeake Ave., Columbus 12. SECRETARY, R. S. Curl, 80 S. Third St., Columbus 15.

• **CINCINNATI:** Organized 1932. Headquarters, Cincinnati, Ohio. Meets, First Tuesday. PRESIDENT, K. B. Little, 832 Temple Bar Bldg., Cincinnati 2. SECRETARY, D. J. Wood, 659 E. Sixth St., Cincinnati.

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• **DELTA:** Organized 1939. Headquarters, New Orleans, La. Meets, Second Tuesday. PRESIDENT, R. B. Guest, 827-37 Dryades St., New Orleans 13. SECRETARY, Theodore Offner, 1000 S. Peters St., New Orleans.

• **GOLDEN GATE:** Organized 1937. Headquarters, San Francisco, Calif. Meets, First Thursday. President, H. V. Hickman, 1129 Folsom St., San Francisco 1. Secretary, R. C. Cushing, 1136 Howard St., San Francisco 3.

• **ILLINOIS:** Organized 1906. Headquarters, Chicago, Ill. Meets, Second Monday. PRESIDENT, J. S. Kearney, 1905 Greenleaf St., Evanston. SECRETARY, G. W. Bornquist, 629 W. Washington Blvd., Chicago 6.

• **INDIANA:** Organized 1943. Headquarters, Indianapolis, Ind. Meets, Fourth Friday. PRESIDENT, W. R. Fenstermaker, 937 Architects & Builders Bldg., Indianapolis 4. SECRETARY, F. C. Barton, 938 K of P Bldg., Indianapolis.

• **IOWA:** Organized 1940. Headquarters, Des Moines, Ia. Meets, Second Tuesday. PRESIDENT, D. E. Wells, 304 Home Federal Bldg., Des Moines 9. SECRETARY, C. P. North, P.O. Box 833, Des Moines 4.

• **KANSAS CITY:** Organized 1917. Headquarters, Kansas City, Mo. Meets, First Monday. PRESIDENT, Henry Nottberg, Jr., 914 Campbell, Kansas City 6. SECRETARY, C. W. Schumacher, 1922 Grand Ave., Kansas City 9.

• **MANITOBA:** Organized 1935. Headquarters, Winnipeg, Man., Canada. Meets, Third Thursday. President, A. W. Moss, 268 Yale Ave., Winnipeg. Secretary, C. M. Fleming, 447 Main St., Winnipeg.

• **MASSACHUSETTS:** Organized 1912. Headquarters, Boston, Mass. Meets, Third Tuesday. PRESIDENT, W. H. Shipp, 124 Lewis Rd., Belmont 78. SECRETARY, J. P. Licandro, 177 State St., Boston.

• **MEMPHIS:** Organized 1944. Headquarters, Memphis, Tenn. Meets, Third Monday. PRESIDENT, W. L. Wellford, Jr., 730 S. Third St., Memphis 2. SECRETARY, A. T. Bevil, 1521 Waverly Ave., Memphis 6.

• **MICHIGAN:** Organized 1916. Headquarters, Detroit, Mich. Meets, First Monday after 10th of month. PRESIDENT, L. A. Burch, 5853 Hamilton Ave., Detroit 2. SECRETARY, J. H. Spurgeon, 5050 Joy Rd., Detroit 4.

• **MINNESOTA:** Organized 1918. Headquarters, Minneapolis, Minn. Meets, First Monday. PRESIDENT, R. C. Jordan, University of Minnesota, Minneapolis 14. SECRETARY, E. F. Snyder, Jr., 4324 Zenith Ave., N., Minneapolis 12.

• **MONTREAL:** Organized 1936. Headquarters, Montreal, Que., Canada. Meets, Third Monday. President, S. W. Salter, 3410 Trenholme Ave., Montreal. Secretary, J. G. Chenevert, 1010 St. Catherine St., W., Montreal.

• **NEBRASKA:** Organized 1940. Headquarters, Omaha, Neb. Meets, Second Tuesday. PRESIDENT, C. A. Carter, 3019 Grebe St., Omaha 12. SECRETARY, K. E. Martin, 203 Patterson Bldg., Omaha 2.

• **NEW YORK:** Organized 1911. Headquarters, New York, N. Y. Meets, Third Monday. President, H. S. Johnson, 39 Cortlandt St., New York 7. Secretary, Carl H. Flink, Room 3000, 51 Madison Ave., New York 10.

• **NORTH CAROLINA:** Organized 1939. Headquarters, Durham, N. C. Meets, Quarterly. President, DeParx Stimson, 121 Idlewilde Dr., Winston-Salem. Secretary, R. B. Crosland, Jr., 1001 S. Marshall St., Winston-Salem.

• **NORTH TEXAS:** Organized 1938. Headquarters, Dallas, Tex. Meets, Third Monday. PRESIDENT, G. A. Linskie, Buckner Blvd., Dallas 10. ACTING SECRETARY, R. G. Lyford, 2415 N. Pearl St., Dallas.

• **NORTHEASTERN OKLAHOMA:** Organized 1948. Headquarters, Tulsa, Okla. President, W. R. Lee, 303 Beacon Bldg., Tulsa 3. Secretary, R. F. Shoemaker, 1136 So. Peoria, Tulsa.

• **NORTHERN OHIO:** Organized 1916. Headquarters, Cleveland, Ohio. Meets, Second Monday. PRESIDENT, R. L. Byers, 1740 E. 12th St., Cleveland 14. SECRETARY, G. V. Parmelee, 94 Solon Rd., Chagrin Falls.

• **OKLAHOMA:** Organized 1935. Headquarters, Oklahoma City, Okla. Meets, Second Monday. President, H. S. Shafer, 400 Key Bldg., Oklahoma City 2. Secretary, R. E. Swan, 117 N.W. First St., Oklahoma City.

• **ONTARIO:** Organized 1922. Headquarters, Toronto, Ont., Canada. Meets, First Monday. PRESIDENT, A. J. Strain, 474 Bathurst St., Toronto. SECRETARY, H. R. Roth, 57 Bloor St., W., Toronto, Ont.

• **OREGON:** Organized 1939. Headquarters, Portland, Ore. Meets, Thursday after First Tuesday. President, J. P. McDermott, 308 Dekum Bldg., Portland 4. Secretary, K. H. Hanson, 6317 S.E. 40th St., Portland.

• **PACIFIC NORTHWEST:** Organized 1928. Headquarters, Seattle, Wash. Meets, Second Tuesday. PRESIDENT, J. D. Sparks, 7331 W. Green Lake Way, Seattle 3. SECRETARY, W. B. Pride, 3228 37th Pl., Seattle 44.

• **PHILADELPHIA:** Organized 1916. Headquarters, Philadelphia, Pa. Meets, Second Thursday. President, F. H. Buzzard, 213-15 S. Broad St., Philadelphia 7. Secretary, C. F. Dietz, 3449 W. Indiana Ave., Philadelphia 32.

• **PITTSBURGH:** Organized 1919. Headquarters, Pittsburgh, Pa. Meets, Second Monday. PRESIDENT, H. J. Kirkendall, 291 Catalpa Pl., Pittsburgh 16. SECRETARY, E. H. Riesmeyer, Jr., 231-33 Water St., Pittsburgh 22.

• **ROCKY MOUNTAIN:** Organized 1944. Headquarters, Denver, Colo. Meets, First Wednesday. PRESIDENT, E. J. McEahern, 1146 Market, Denver 4. SECRETARY, N. H. Brickham, 2715 S. Humboldt, Denver.

• **ST. LOUIS:** Organized 1918. Headquarters, St. Louis, Mo. Meets, First Tuesday. PRESIDENT, W. A. Russell, 7918 Kingsbury Blvd., Clayton 5. SECRETARY, H. C. Sharp, 4060 W. Pine Blvd., St. Louis 8.

• **SHREVEPORT:** Organized 1948. Headquarters, Shreveport, La. Meets, Third Thursday. President, A. H. Otto, 1020 Market St., Shreveport. Secretary, S. W. Beaty, P.O. Box 1734, Shreveport.

• **SOUTH TEXAS:** Organized 1938. Headquarters, Houston, Tex. Meets, Third Friday. PRESIDENT, R. J. Salinger, 4510 Dewberry Lane, Houston 4. SECRETARY, G. J. Collins, P.O. Box 2463, Houston.

• **SOUTHERN CALIFORNIA:** Organized 1930. Headquarters, Los Angeles, Calif. Meets, Third Wednesday. PRESIDENT, J. L. Blake, 1700 S. Bedford St., Los Angeles 24. SECRETARY, M. C. Greiner, 1559 Club View Dr., Los Angeles 24.

• **SOUTHWEST TEXAS:** Organized 1946. Headquarters, San Antonio, Tex. Meets, Second Tuesday. President, G. R. Rhine, 1208 Alamo National Bldg., San Antonio. Secretary, I. W. Wilke, P.O. Box 1154, Austin 3.

• **UTAH:** Organized 1944. Headquarters, Salt Lake City, Utah. Meets, First Wednesday. President, C. E. Ferguson, 838 E. Garfield Ave., Salt Lake City. Secretary, A. R. Curtis, 2211 Highland Dr., Salt Lake City.

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• **WESTERN MICHIGAN:** Organized 1931. Headquarters, Grand Rapids, Mich. Meets, Second Monday. PRESIDENT, J. A. Alexander, P. O. Box 132, Station C, Grand Rapids. SECRETARY, L. A. Calcaterra, 1136-42 Plainfield Ave. N.E., Grand Rapids.

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• **WISCONSIN:** Organized 1922. Headquarters, Milwaukee, Wis. Meets, Third Monday. PRESIDENT, B. M. Kluge, 1817 So. 66th St., Milwaukee 14. SECRETARY, J. A. Lofte, 9121 W. Hawthorne Ave., Milwaukee 13.

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• **LOUISIANA POLYTECHNIC INSTITUTE:** Organized 1949. Headquarters, Ruston, La. PRESIDENT, J. C. Cox, 417 W. Mississippi, Ruston. SECRETARY, H. D. Hayes, Box 409 Tech Station, Ruston.

• **NORTH CAROLINA STATE COLLEGE:** Organized 1948. Headquarters, Raleigh, N. C. PRESIDENT, J. J. Andrews, 710 Harvey St., Raleigh. SECRETARY, E. A. Stroupe, Jr., 416 Halifax St., Raleigh.

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• **TEXAS A. & M. COLLEGE:** Organized 1946. Headquarters, College Station, Tex. Meets, Second and Fourth Tuesday. PRESIDENT, R. N. Hughes, Box 914, College Station, Tex. SECRETARY, W. J. Bowen, Box 1913, College Station, Tex.

• **UNIVERSITY OF DETROIT:** Organized 1949. Headquarters, Detroit, Mich. Meets First Tuesday. PRESIDENT, L. M. Blanchette, 2245 Holcomb, Detroit 14. SECRETARY, S. H. Kosinski, 4439-30th, Detroit 10.

• **UNIVERSITY OF KANSAS:** Organized 1949. Headquarters, Lawrence, Kans. President, V. N. Elder, 643 Tennessee, Lawrence. Secretary, L. H. Noll, 1235 New York St., Lawrence.

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EQUIPMENT DEVELOPMENTS

For your convenience in obtaining more information about any of this equipment, see coupon on this page. Add the new products and companies listed here to your Directory Section which you received in your January 1949 Heating, Piping & Air Conditioning and thus keep your records of sources of supply up to date throughout the year. Single asterisk indicates equipment not listed in Directory Section; double asterisk equipment and manufacturer not listed.

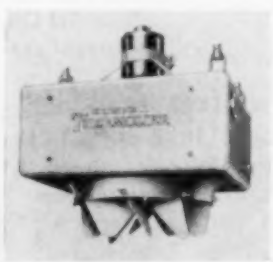
Compressible Lubricant Provides Automatic Lubrication of Plug Valves

No. 4181—"Hypermatic" is the name of a new plastic lubricant which is both compressible and expansible. Developed by Nordstrom Valve Div., Rockwell Mfg. Co., 400 N. Lexington Ave., Pittsburgh 8, it is said to provide vast savings in maintenance and servicing costs of pressure lubricated valves.

Compression of the lubricant in the valve storage reservoir creates an "energized" condition which, the manufacturer claims, automatically feeds the lubricant to seal seating surfaces and to fill voids in the event of a leak. Automatic lubrication is said to exist until the confined pressure drops to 50 psi. Moreover, says the company, adequate pressure is maintained for an almost "limitless number of valve turnings." When the pressure does drop below 50 psi, automatic operation may be restored by adding more lubricant or by turning the lubricating screw to create pressure. Should the valve be neglected over a long period and the energy dissipated, the material reverts to the status of a nonautomatic lubricant.

While it was developed primarily for pressurized plug valves, the company believes it will have numerous applications in other fields. It is available in stick or bulk form. In the illustration a stick of the lubricant is being inserted and the gage demonstrates pressure built up.

Plastic Lubricant



Unit Heater

Three Point Suspension and Internal Cooling Leg Feature Vertical Unit Heater

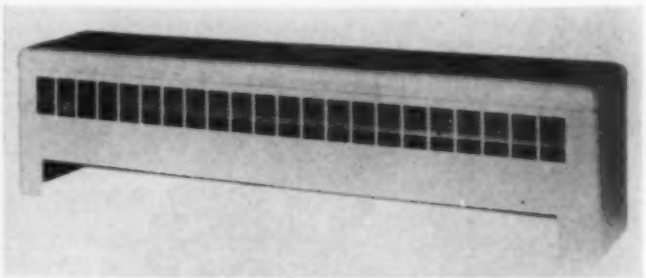
No. 4182—A new "Thermolier" unit heater for vertical delivery of heated air from heights of 9 to 25 ft above the floor has been developed by Grinnell Co., Inc., 277 W. Exchange St., Providence 1, R. I.

Three point suspension from adjustable swivel couplings is used to simplify suspension and alignment. Supply and return piping connections are on one side of the unit, and the patented internal cooling leg and close connection of a thermostatic trap are said to insure continuous drainage and full capacity under all conditions of use. Each individual finned U tube is pitched for continuous drainage.

Six models are available in capacities from 50,800 to 257,000 Btu per hr (based on 2 psi steam and 60 F entering air).

Convector-Radiator Fits Under Picture Windows

No. 4183—The Trane Co., La Crosse, Wis., has introduced a new convector-radiator especially designed for installation under picture windows.



Three basic models are available; free standing, semi-recessed, and fully recessed, in lengths up to 88 in. The units are 12 in. high and are offered in 4, 6, 8, and 10 in. depths. The free standing unit is designated as model FKL and model RKL is for recessed installation. Both may be used with either steam or hot water.

Induction Motor Has Stator Frame of Rolled Steel

No. 4184—New "Life-Line" a-c induction motors, ranging from 20 to 75 hp, are being produced by Westinghouse Electric Corp., P.O. Box 868, Pittsburgh 30.

The stator frame is made of solid heavy rolled steel. To control the inner diameter accurately enough to accommodate the stator laminations, the frame is rolled from flat steel and welded into a ring while on a mandrel. The solid frame is not only said to give unusual protection but to permit wall or ceiling mounting with equal protection. Brackets or end bells are drip-proof, with the top half solid, and ventilating openings on the lower half only. Both sleeve and ball bearing brackets are available.

The motors are offered in the squirrel cage and wound rotor types, are both drip-proof and splashproof, and are available in 60, 50, and 25 cycles.

[*Continued on page 188*]

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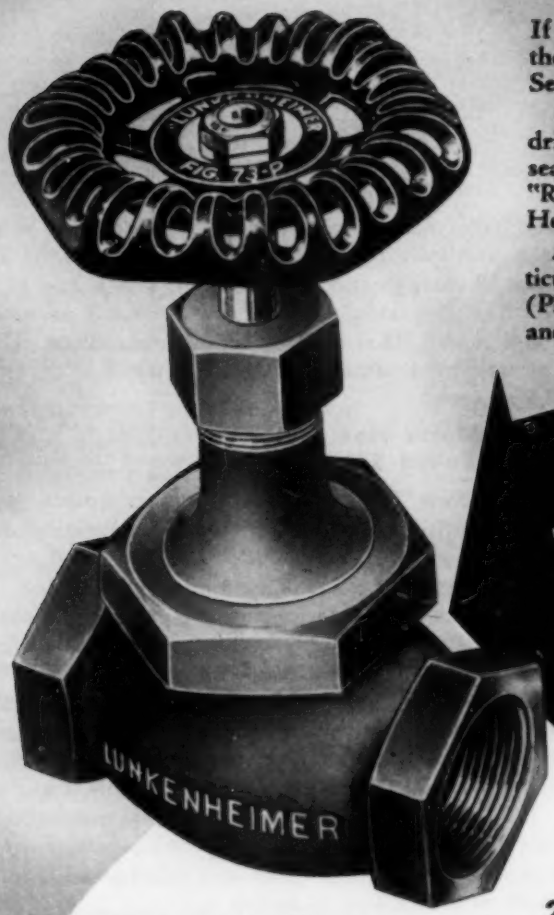
Here's a Tip . . .

FOR SELECTING THE *RIGHT VALVE* FOR THE *RIGHT PLACE!*



"Renewo" Globe Valve Regular Type, "P" Type or "PS" Type

If you just want an extra good valve for general service, then choose the "Renewo" with Regular Type Nickel Alloy Seat and Disc. It's tops in its class.

Or, perhaps you have a throttling condition such as drain, drip, water column blow-down and like service where regular seats and discs tend to cut out . . . use the "P" (Plug Type) "Renewo" with 330 Brinell "NS 5" Alloy Seat and Disc. Here you have high resistance to wear and corrosion.

And for severe close throttling and erosive service, particularly where abrasive conditions are present . . . the "PS" (Plug Type) "Renewo" with 500 Brinell Stainless Steel Seat and Disc will solve the problem.

LUNKENHEIMER "RENEWO" VALVE *Threesome*

and look at these money saving extra features

1. REGULAR TYPE *"Renewo"* Nickel Alloy Seat and Disc for general service.

2. "P" (PLUGTYPE) *"Renewo"* "NS5" Nickel Alloy 330 Brinell Seat and Disc for throttling, drain, drip, water column blow-down and like service.

3. "PS" (PLUG TYPE) *"Renewo"* Stainless Steel 500 Brinell Seat and Disc for close throttling and other severe erosive service.

1. Each type of seat and disc is interchangeable one with another, making it possible to convert a regular type valve into a plug type without taking the valve out of the line.

2. Available in 200 lb. S.P. as well as 300 lb. S.P. making it unnecessary to purchase a 300 lb. valve for 200 lb. service.

The "Renewo" Threesome is fully described in Circular No. 577. A copy is yours for the asking . . . from your Lunkenheimer Distributor or direct from us.

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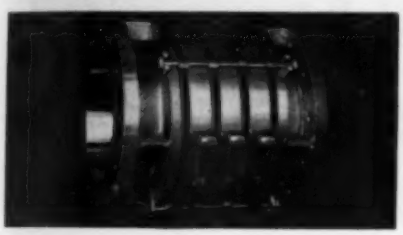
Zallea Expansion Joint News

PUBLISHED BY ZALLEA BROTHERS—WORLD'S LARGEST MANUFACTURERS OF EXPANSION JOINTS EXCLUSIVELY

Expansion Joint of the Month

SELF-EQUALIZING TYPE

When the traverse is more than ½", or pressure greater than 30 lbs., several corrugations are necessary, but no matter how perfectly the corrugations are formed, several may be more easily compressed than others. Where an expansion joint must absorb considerable movement, it is mandatory that no corrugation be compressed or opened more than the proportionate amount of movement, for which it is designed. Therefore, the equalizing rings on this Zallea Expansion Joint serve to distribute the compression equally among all the corrugations. The rings which surround the corrugations have a shape corresponding to the compressed corrugation. When normally installed, the rings are separated from



each other by a space equal to the traverse which each corrugation can absorb. When an expanding pipe line exerts a thrust against the end fittings of the expansion joint, each corrugation is partly compressed, but as soon as one corrugation has taken up its share of the movement, the rings on each side bear upon each other, and prevent that corrugation from being further compressed; all the thrust being transmitted to the other corrugations through the equalizing rings. The rings also support the element against internal pressure.

. . . a complete, practical catalog on all Zallea Expansion Joints. A text book and a valuable reference source. Send for your *free* copy *today!*

ZALLEA BROTHERS, 816 Locust Street, Wilmington 99, Delaware.

HOT SPRINGS HEAT THE ARCTIC!

Control in Manufacture

THE EXPANSION PROBLEM

One of the most important problems facing the designer of a power or process piping system is the selection of the best means of absorbing the expansion and contraction due to temperature changes or external stresses. The selection may be considerably complicated by extremes of corrosion, temperatures, and pressure. Generally speaking, one of the Zallea types of corrugated expansion joints, when properly selected and installed, is the best means of compensating for expansion and contraction of pipe lines in sizes 3" diameter and larger; for pressures up to 1000 psi. and for temperatures up to the 1600° F. Metals suited to most chemical services are available for use in the Zallea corrugated expansion element. There is a Zallea Expansion Joint for every service: the non-equalizing type for pressures up to 30 lbs. and traverse up to ½"; the self-equalizing type for pressures up to 300 lbs. and traverse up to 7½" in a single unit; the duo-equalizing type for the same service as the self-equalizing type, but better suited for frequent movement; the flexible connector for combined axial and lateral movement at atmospheric pressure; the Universal type for combined axial and lateral movement for pressure applications.

NATURAL PHENOMENON PROVIDES HOT WATER

Strange as it may seem, one of the Arctic's largest cities obtains its hot water from an underground labyrinth of hot springs! This natural phenomenon provides Reykjavik, Iceland with one of the most unusual heating sources. The hot water is pumped from the springs through a series of underground pipe lines fitted with Zallea Corrugated Self-Equalizing Expansion Joints.

Literally Buried Alive

Under tons of rock and dirt these Zallea Expansion Joints are at work, absorbing thermal expansion and contraction of the pipe lines. These moving, breathing expansion joints are literally "buried alive." Shown below is a welder installing one of over 400 Zallea Expansion Joints in what is recognized to be the world's largest hot water district heating system. These Self-Equalizing Expansion Joints are particularly suited to this installation because there is no packing to deteriorate and no maintenance is required.

ELCO

1 FAST ASSEMBLY

2 TENSILE STRENGTH

3 UNIFORMITY

all 3 ARE IMPORTANT

IN SELLING YOUR PRODUCT

Fast assembly means lower costs and reduced selling prices. Elco quick starting points and uniform heads and slots make assembly easier and faster.

Constant holding power with high rattle-resistance comes from analyzed steel used in all Elco fasteners. This means a dependable product, free from maintenance.

Uniformity to a fine degree is the result of Elco rigid manufacturing control. This is assurance against defects and costly assembly delays.

The dependability of Elco fasteners in providing fast assembly, maximum tensile strength and high uniformity results in a superior product without extra cost. Write today and inquire about Elco fasteners to meet your requirements.

ELCO Screws

ELCO TOOL AND SCREW CORPORATION

1920 BROADWAY ROCKFORD, ILLINOIS

WOOD SCREWS • MACHINE SCREWS • MACHINE SCREW NUTS
DRIVE SCREWS • CAP SCREWS • LAG SCREWS • SPECIAL SCREWS
TAPPING SCREWS • STOVE BOLTS • PIPE PLUGS

[Continued from page 185]

New Diffuser Handles Supply and Return Air

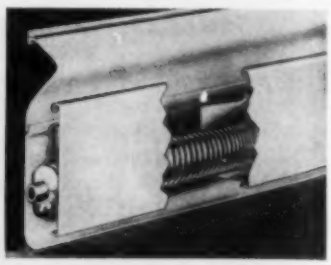
No. 4185—Attractive styling, adjustable air pattern, and high air diffusion efficiency are said to characterize the new "Venturi-Flo" air diffusers of Barber-Colman Co., Rockford, Ill.

In the model JC combination supply and return unit return air volume is regulated by a key adjustment. It is said to be well suited to installations where the structure will not permit returns in the side walls.

The model J supply unit is similar in appearance, and a screw in the center of the diffuser provides adjustment of the supply air pattern from horizontal to vertical deflection. According to the company, rapid temperature equalization and draftless air distribution are provided by the venturi design which mixes room air and supply air at the outlet.

Both models are offered in a wide range of sizes and finishes.

Air Diffuser



Baseboard Heating Unit

Baseboard Heating Unit Has Copper Tubing, Helical Fins

No. 4186—"National Art" baseboard heating units are now being marketed by National Radiator Co., Johnstown, Pa.

The heating element is made of ¾ in. commercial copper tubing on which is bonded a series of helical copper fins. The steel enclosure is provided in the form of a back panel assembly and a front panel assembly and units are packaged so that the back panel and heating element may be installed along the wall surface without opening front panel cartons until needed for job completion.

Hot water ratings range from 370 Btu per hr per lineal ft at a water temperature averaging 170 F up to 730 Btu per hr per lineal ft with a water temperature averaging 230 deg.

Instrument Valves Provide Close Regulation on Small Lines

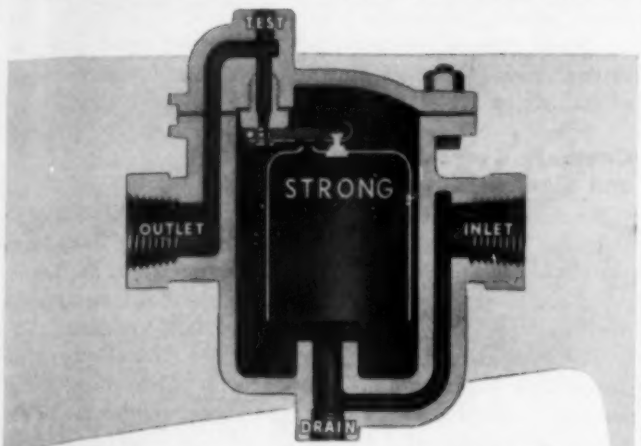
No. 4187—New forged steel instrument valves now being built by Edward Valves, Inc., East Chicago, Ind., are said to be exceptionally strong and easy to install in a limited space. They are particularly recommended for use with meters, gages, pumps, regulators, and other small control equipment which is accompanied by an instrument valve.

Of carbon steel, 13 percent chromium stainless, or 18-8 stainless steel, these valves are now available in globe or angle

UNIVERSITY OF MICHIGAN LIBRARIES

for small drainage jobs . . .

A NEW *SMALL* TRAP WITH *BIG* TRAP QUALITY FEATURES!



- Renewable seat—easily removed for servicing.
- Strong's patented, wear-resistant Anum-Metl† valve and seat—guaranteed leak-proof for one year. Scientifically designed Hi-Cap* orifice gives greater capacity.
- Oversized parts—increased capacity.
- Horizontal inline connections—simplifies piping.
- Stainless 18-8 high ratio leverage.
- Large gasket area.
- Seamless, deep drawn, stainless 18-8 bucket—no welds.
- Bottom drain or inlet connection.

Available for pressures from 0 to 150 p.s.i., the Strong* No. 070 fits all standard trap applications. Built especially for use on laundry, restaurant and hospital equipment and in industry wherever small drainage is a problem.

As with all Strong Semi-Steel Traps, the 070 is guaranteed against defects in material and workmanship for one year. Write today *for more complete information.*

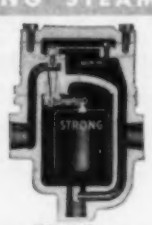
*T. M. Reg. U. S. Pat. Off.

STRONG, CARLISLE & HAMMOND COMPANY
1392 West 3rd Street
Cleveland 13, Ohio

† ANUM-METL Reg. Trade Mark

OTHER STRONG STEAM SPECIALTIES

Reducing Valve



Blast Trap

Strainer

STRONG

ONLY *Combustioneer* Has The *Automatic Respirator* WHICH CUTS FUEL COSTS UP TO 30%

Commercial Model No. 50

Combustioneer's Automatic Respirator, exclusive and patented, maintains at all times the precisely correct amount of air in the fire-bed for ideal combustion. There are no dampers or shutters in the air duct to adjust—operation is completely automatic. This assures maximum efficiency . . . *and cuts fuel costs up to 30%. Only* Combustioneer has it!

Combustioneer's exclusive "Pulsating" Transmission agitates the fire-bed, thoroughly mixing air and coal so that every particle is consumed. There is no waste. Its direct gear contact gives tremendous power. Its heavy-duty ruggedness assures long trouble-free service. Feed rate changes can be made easily while the stoker is in operation.

Twenty years of precision-engineering leadership have produced a complete line of hopper and bin-feed Combustioneers with capacities up to 1000 pounds per hour. They meet all heating needs in factories, shops, stores, theaters, hotels, apartment houses and other large private and public buildings.

Write today for the proof that a Combustioneer Automatic Coal Stoker can *cut your fuel costs!*

The Steel Products Engineering Co.
1238 W. Columbia St., Springfield, Ohio.

Please send us, without obligation, further information about your commercial models of Combustioneer, and information about your free heating analysis.

NAME_____

KIND OF BUSINESS_____

COMPANY_____

ADDRESS_____

CITY_____ ZONE___ STATE_____

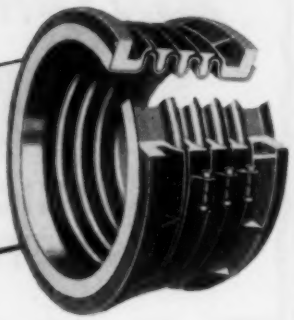
ANOTHER REASON WHY

Badger

IS FIRST IN

expansion joints

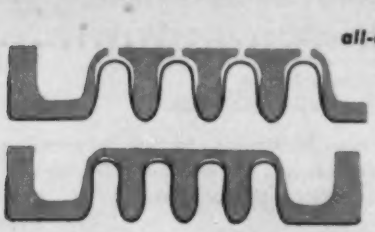
COMPACT



Installation and maintenance in tight places are easy when you use Badger Packless Corrugated Expansion Joints. Thanks to thoughtful designing, the outside diameter of any Badger Joint is about the same as any ordinary flange. It's one reason why users everywhere insist on the *original* Badger Packless Corrugated Joint. Here are other features equally important to you:

- Flexible . . . quick response to heat expansion cuts strain on adjacent fittings and equipment
- Packless . . . single tube construction provides pressure-tight joint without packing
- Special forming methods . . . no danger of structure-weakening stresses
- Controlled heat treating . . . to assure long life for flexing member
- Directed flexing rings . . . to equalize stresses *between* and *within* corrugations
- Fabricated from stainless steel or alloys . . . joints withstand high temperature and corrosion

These features are important in any pipe line job — you can get them all only by specifying Badger Expansion Joints.



Badger

MANUFACTURING COMPANY

230-260 BENT ST. CAMBRIDGE 41, MASS.

design in ¼ in., ⅜ in., and ½ in. sizes with screwed or socket welding ends. They have outside screw and yoke construction and the yoke is screwed onto the body and tack welded. The carbon steel valves are rated at 1500 psi at 850 F and the 13 percent chromium and 18-8 stainless steel designs are rated at 1500 psi at 1000 F. The WOG rating for all of these valves is 6000 psi at 100 F.

They are said to be ideal for providing close regulation on orifice meters, regulator lines, gage lines, instrument panels, Christmas tree valve constructions in the oil fields, pumps, bypass lines, and other small lines in central power stations and boiler houses of many types of industries.

Combats Corrosion in Condenser and Cooling Tower Sump Tanks

No. 4188—To combat electrolytic corrosion of zinc galvanizing on sump tanks, the "Lektor-Tektor" has been developed by Marlo Coil Co., 6135 Manchester, St. Louis, Mo., and is now being installed as standard equipment on its evaporative condensers and cooling towers.

The new device is a "sacrificial" rod installed in the sump tank slightly below water level. According to the company, electrons produced by the rod render harmless dissolved oxygen and other corrosive elements. A protective film is also said to be formed on the tank walls as the continuous flow of current results in deposits of insoluble lime salts on the walls, and the tank water is chemically conditioned to render it less aggressive. Magnesium is introduced into the water from the rod, and this is said to strengthen the protective film.

Protected Against Corrosion

Air Cleaner, Humidifier

Humidifies, Cleans, and Purifies Air

No. 4189*—The series 1700 horizontal or ceiling type "Air-Renewer" is offered by Brundage Co., Kalamazoo, Mich., to humidify, clean, and purify air by a glycol filter compartment and germicidal lamps.

Made in two sizes, 1200 and 2400 cfm, the unit combines glycol scrubbing and ultraviolet radiation. Heating or cooling coils may be installed with the unit, and according to the company, the series 1700 lends itself to conditioning of larger spaces by installation in banks of two or more combined with a central blower installation.

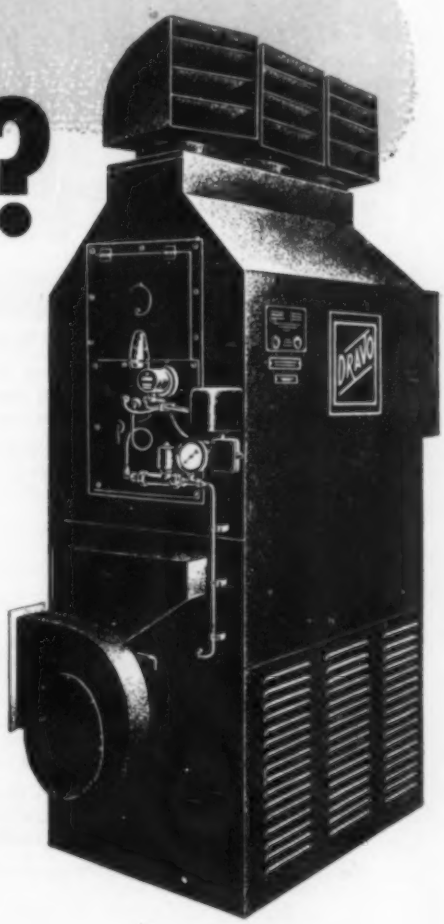
Coupling Has All-Rubber Center

No. 4190—The "Mighty Midget," an all-rubber-center coupling designed for improved power transmission in fractional horsepower motors, has been added to the line of "Morflex" couplings produced by Morse Chain Co., 7601 Central Ave., Detroit 8, a subsidiary of Borg-Warner Corp.

MOST HEATING SYSTEMS END UP BY

WARMING THE AIR

WHY NOT BEGIN WITH IT?



DRAVO *Counterflo* HEATERS

Dravo *Counterflo* Heaters provide a highly efficient heating method for open area structures. This method is efficient because it heats the air right where it's to be used and without employing water or steam as intermediaries.

No "conversion" losses with Dravo *Counterflo* Heaters on the job. The heater warms the air, and the positive, directed circulation floods the working zone with warmth. Burns oil or gas as fuel—can be readily converted from one to the other. Low fuel cost—80 to 85% efficiency. Low installed cost—users report 50 to 66% less investment than in standard wet type heating systems. Immediate delivery, quick, easy installation, no delays due to pipe shortages. Thousands of successful installations. Write for Bulletin CD-523-354.

DRAVO
CORPORATION
DRAVO BUILDING, PITTSBURGH 22, PA.

Dravo also manufactures the *DRAVO CRANE CAB COOLER* for air conditioning hot-metal crane cabs.

PITTSBURGH • CLEVELAND • PHILADELPHIA • DETROIT • NEW YORK • CHICAGO • ATLANTA • BOSTON

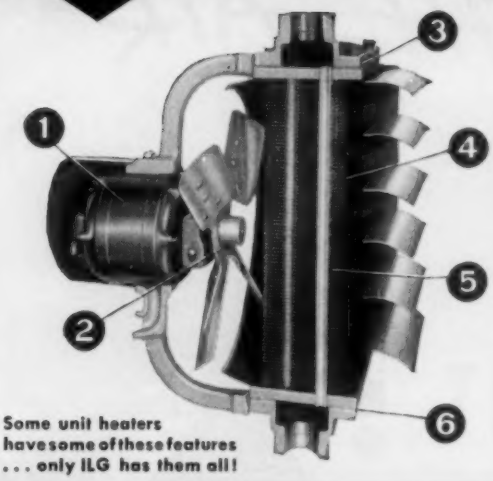
Sales Representatives in Principal Cities. Mfd. and Sold in Canada by Marine Industries, Ltd., Sorel, Quebec.

DRAVO

THE WORLD'S FINEST UNIT HEATER!

ILG

Extra quiet! Extra capacity! Extra economy!



Some unit heaters have some of these features . . . only ILG has them all!

One-name-plate . . 2-piece header . . No brazed joints

1. Patented ILG-built Self-Cooled Motor—never "slow roasts" due to radiated heat from core . . . saves 5% to 10% on power cost.
2. Dynamically balanced fan wheel—specifically designed for ILG Unit Heaters . . . verified for dynamic balance at rated speeds.
3. Two-piece header bolted together—graduated for "balanced" steam distribution. Trouble-free operation on steam pressures to 125 lbs.
4. Solid copper core . . . 26 gauge flanged copper fins pressed on to ½" O.D. seamless copper tubes. ILG *never* has substituted less efficient, less durable metals in its fins or tubes.
5. No welded, soldered or brazed joints. Pressing operation *permanently* unites fins and tubes. Brass orifice bushings driven into entrance of tubes to expand tubes uniformly into header.
6. Bottom Header—"Floats" on slotted coil retainer and spring washers—coil expands and contracts independent of unit casing.

PLUS the famous ILG "One-Name-Plate" Responsibility covering both unit and motor.

Sell nationally-advertised ILG Unit Heaters. Only ILG has all these features . . . only ILG offers so much value per dollar.

Send coupon today for FREE Unit Heater Bulletin 342

ILG Electric Ventilating Co.,
2841 N. Crawford Ave., Chicago 41, Ill.
Offices in more than 40 Principal Cities

Firm Name _____

Your Name _____ *Title* _____

Address _____ *Zone* _____

City _____ *State* _____

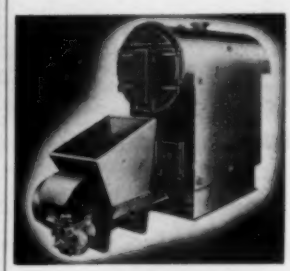
The neoprene center member is designed to give the new coupling an unusually long service life. Flexing caused by misalignment is said to be absorbed by the entire center member and the tight fit of the coupling pins, when pressed into the rubber center member, is used to eliminate noise and vibration.

Among the many applications stressed by the manufacturer are pumps and fans.

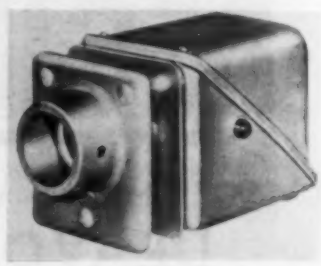
Preinstallation of Hearth and Retort Features Boiler-Stoker Combination

No. 4191—"Unitbuilt" boiler-stoker combinations in 10 sizes, with steam radiation capacities of 2190 to 8500 sq ft, are available from Brownell Co., 408 N. Findlay St., Dayton 1, Ohio.

The stoker ironwork is factory installed in the boiler furnace. After the boiler has been set on location, the stoker driving mechanism is connected. Preinstallation of the hearth and retort is said to greatly simplify the final assembly work and to reduce the installation time and expense. Cleanout doors are provided in the front and rear of the base. Both stoker and boiler are company products.



Boiler-Stoker Unit



Oil Burner Control

Protects Commercial, Industrial Oil Burners Against Flame Failure

No. 4192—"Fireye" type F18T-2 consists of a phototube and amplifying system in an aluminum case and is designed to provide operating flame failure protection for manually ignited oil burners, air heaters, etc. and is used with programming controls to protect automatically ignited commercial and industrial oil burner installations. The manufacturer is Combustion Control Corp., 77 Broadway, Cambridge 42, Mass.

Special heat resisting filters and heat insulating fins and spacers are used so that the entire control may be mounted directly on the furnace wall. A hermetically sealed time delay unit is used to prevent the relay from dropping out from purely transitory flame disturbances, and a pilot light indicating flame failure can be viewed through an angle of 180 deg.

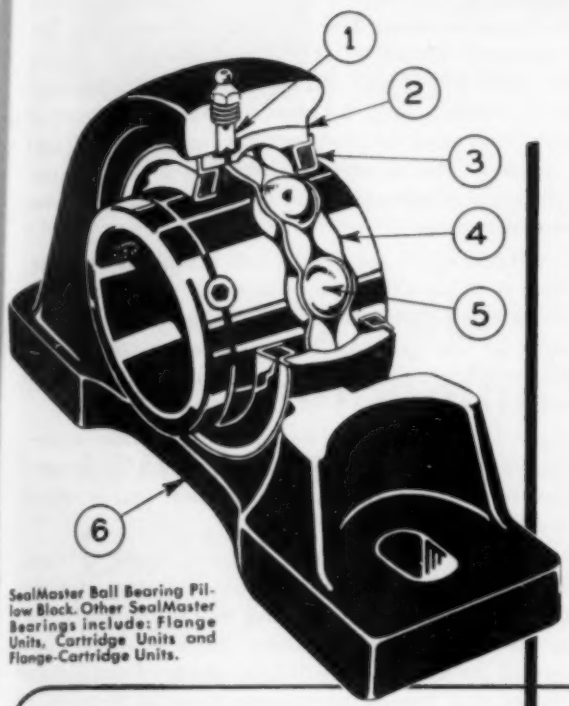
Refrigeration Relief Valve Has Replaceable Rupture Disc

No. 4193—The new type 545 diaphragm relief valve, offered by Henry Valve Co., Melrose Park, Ill., is designed for use with "Freon" and has a built-in replaceable rupture disc.

Factory tests are said to show that it will operate with a deviation from set pressure considerably less than 5 percent permitted under the ASA-B9 code. It will be demonstrated this month at the 6th All Industry Refrigeration and Air Conditioning Exposition.

Design *Your* fans, blowers, and air-conditioning equipment to include SEALMASTER

BALL BEARING UNITS



SealMaster Ball Bearing Pillow Block. Other SealMaster Bearings include: Flange Units, Cartridge Units and Flange-Cartridge Units.

1. NO HOUSING WEAR!
Patented locking pin and dimple prevent rotation of outer race in housing, thus eliminating housing wear . . . permit shaft alignment and position unit for relubrication.

2. SELF-ALIGNING!
Bearing unit, with seals independent of housing, can align itself in any direction without seal distortion.

3. PERMANENTLY SEALED!
Felt-lined steel flinger rotating in labyrinth prevents entry of dirt and retains proper amount of lubricant.

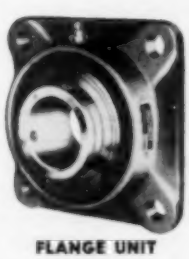
4. FLOATING RETAINER!
Ball retainer is designed to float on ground inner surface of outer race . . . traps lubricant and prevents churning.

5. PRE-LUBRICATED!
The bearing chamber is filled with the proper amount of grease before bearing leaves factory.

6. HOUSING DESIGN!
SealMaster's cast housings combine rigid one-piece construction with neat streamlined appearance.

SILENCE Your Air-Conditioning Equipment with These SEALMASTER UNITS

4-BOLT BASE PILLOW BLOCK



FLANGE UNIT

STANDARD DUTY PILLOW BLOCK

A complete line of SealMaster pillow blocks, flange units and other types are available in a wide range of shaft sizes and for varying degrees of service. SealMaster units are designed and constructed to give top performance on all types of fans, blowers, pumps and other equipment in the air conditioning field. Include SealMaster bearings in your design and improve your product. Write us for a free copy of SealMaster catalog No. 845.

Climate changer unit for commercial and residential air conditioning uses SealMaster bearings for noise-free, dependable operation. Trane Company photograph.

BEARING DIVISION

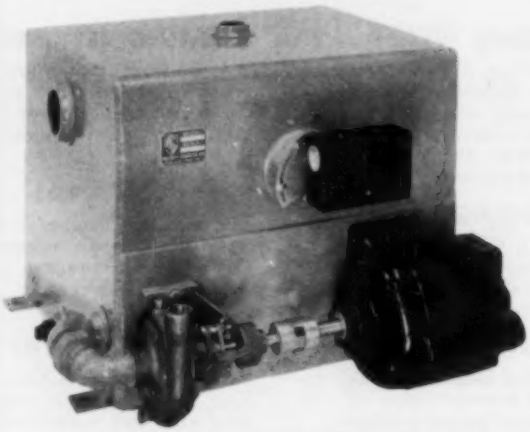
STEPHEN S-A ADAMSON
MFG. CO.

47 RIDGEWAY AVENUE, AURORA, ILLINOIS
LOS ANGELES, CALIF. ★ BELLEVILLE, ONT.

Factory Representatives and Dealers in All Principal Cities

HOT NEWS!

Announcing THE EAGAN "EAGLE" Condensate Pump



MATCH THIS FOR VALUE!

Here's the best condensate pump "buy" on today's market. It provides twice the capacity of any similar pump . . . . 8,000 sq. ft. EDR at 20 PSI . . . . and it's only $124.00 (trade price).

COMPARE THESE FEATURES

- Low inlet, 2" IPS, 12" from floor.
- All bronze centrifugal pump.
- Flexible coupling.
- Standard 1/3 HP capacity motor.
- Thermal overload protection.
- Level indicating 2 pole float switch.

WALTER H. EAGAN CO., INC.
Mfgrs. of Condensate, Vacuum, Turbine, Boiler Feed and Centrifugal Pumps.
Pump Specialists Since 1920

CLIP COUPON *for Free Bulletin*

WALTER H. EAGAN CO., Inc. (Dept. A)
2336 Fairmount Avenue
Philadelphia, 30, Pa.

SEND ME FREE DESCRIPTIVE BULLETIN OF THE NEW EAGAN "EAGLE" CONDENSATE PUMP.

NAME.....

COMPANY.....

STREET.....

CITY..... ZONE.... STATE.....

Provides Independent Control of Temperature and Humidity

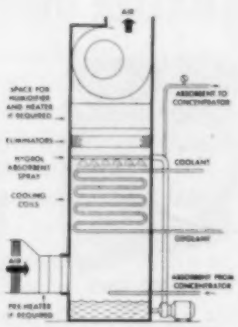
No. 4194—A new air conditioning unit designed to give complete control of temperature and relative humidity is being offered by Niagara Blower Co., 405 Lexington Ave., New York 17. It is said to provide dry air at normal atmospheric temperatures with little or no refrigeration required.

The apparatus is enclosed in a casing through which the air is drawn by fans. The air is filtered and then enters a chamber where it is dehumidified in passing through a spray of "Hygol" liquid. In the same chamber are located cooling coils to remove the latent heat of evaporation and also do any sensible heat removal that may be needed. The air stream then passes through entrained moisture eliminators and is diffused in the space to be conditioned.

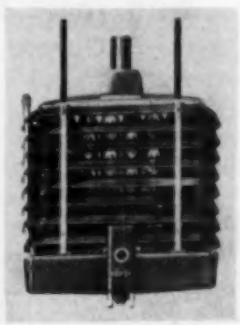
The absorbent liquid spray falls into a tank at the base, from which it is piped to a concentrator which removes the moisture taken from the air. The reconcentrated liquid returns to the system. This process is continuous, and the apparatus is designed to operate at full capacity at all times.

The same equipment may be used to provide winter air conditioning when required, by installing a tempering coil at the outdoor intake, a humidifier, and a reheat coil above the eliminators.

This equipment is manufactured in a range of sizes from 1000 to 20,000 cfm, and multiple unit installations are said to be practical. According to the company, by reducing the need for refrigeration, the cost of air conditioning will be reduced by this method. Applications generally are in a temperature range from 35 F upward. Below the freezing temperature of water, the firm's "No Frost" method is recommended.



Air Conditioner



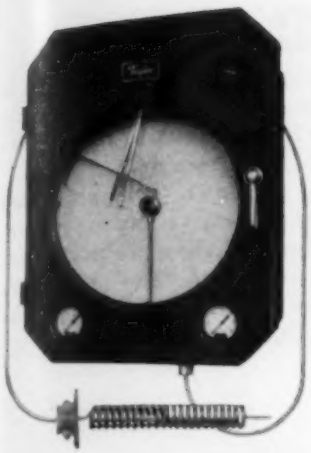
Gas Heater

Suspended Gas Fired Heater Provides Radiant Heat

No. 4195*—The "Overhead Panelray" gas fired heating unit has been designed to utilize infrared rays and to provide instantaneous heat that warms occupants and solid objects regardless of air temperature. The manufacturer is Day & Night Div., Affiliated Gas Equipment, Inc., Monrovia, Calif.

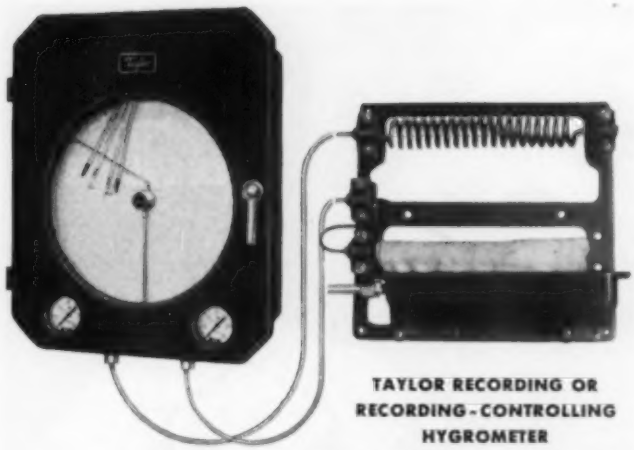
Completely automatic and AGA approved, the heater is suspended, does not utilize a fan or blower, and is said to direct radiant heat downward by means of a series of louvers.

It is said to offer many advantages in applications such as warehouses, factories, garages, stores, etc.



TAYLOR FULSCOPE* TEMPERATURE CONTROLLER

Regulates diaphragm valves and lever motors for operation of dampers, etc., to make your air conditioning system work smoothly. Automatically compensates for load changes. Controls refrigeration compressors. Gives you automatic seasonal changeover.



TAYLOR RECORDING OR RECORDING-CONTROLLING HYGROMETER

With continuous water feed. Ideal where air circulation is good, and panel is well located within apparatus, room or duct for best circulation across bulbs. Available with constant level or bottle-fed water reservoir.

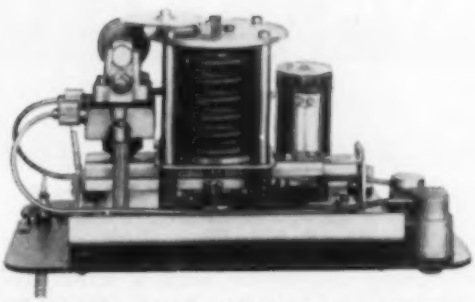
What's *your* air conditioning control problem?

DO you need a single control instrument for an isolated application? Or complete controls for a complete air conditioning system?

Taylor makes a full line of accurate control instruments for every air conditioning need. We'll be glad to work with you or your consulting engineers or contractors to give you the instruments best suited to your individual problem. Ask your Taylor Field Engineer or write Taylor Instrument Companies, Rochester, N. Y., or Toronto, Canada.

TAYLOR FULSCOPE* RATIO RECORDING CONTROLLER

Ideal where you need control in one medium in a definite relationship to some other variable medium . . . such as inside temperature control in summer air conditioning according to variable outdoor temperatures.

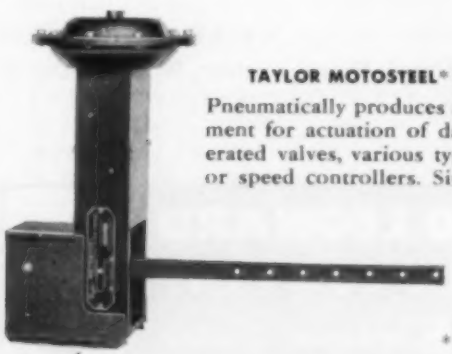


TAYLOR TRANSAIRE* TEMPERATURE TRANSMITTER

New force-balance pneumatic transmitter that meets all normal air conditioning requirements with Simple Operation, Positive Transmitting action and Rapid Response to Temperature Changes. Ideal where zones are remotely located from centralized control room.

TAYLOR 108G HYGROMETER

With bottle-fed reservoir. Perfect for more accurate spot checks on air conditioning. Combines accuracy of an etched stem thermometer with ruggedness of an industrial thermometer. Easier to read, install and maintain. Handles jobs like: air conditioning supply and return ducts, dryers, and other closed compartments where you want accurate temperature and humidity readings.



TAYLOR MOTOSTEEL* LEVER MOTOR

Pneumatically produces a thrusting movement for actuation of dampers, lever-operated valves, various types of governors or speed controllers. Simple, low-priced.

* Registered Trade-Mark

TAYLOR INSTRUMENTS MEAN ACCURACY FIRST

because contacts get a GENEROUS WIPE

Unique spring suspension adds a self-cleaning action to the heavy, silver-to-silver contacts of Ward Leonard's 105 Heavy-Duty Midget Relays. These features make them self-aligning, too.

Use Ward Leonard 105 Midgets on jobs normally requiring heavier relays.

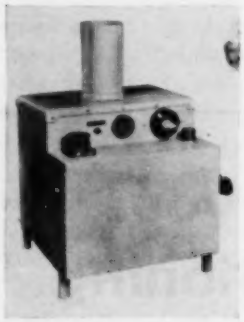
Write for Relay Catalog. Ward Leonard Electric Co., 24 South Street, Mount Vernon, N. Y. Offices in principal cities of U. S. and Canada.

WARD LEONARD ELECTRIC COMPANY
Result-Engineered Controls
RESISTORS • RHEOSTATS • RELAYS • CONTROL DEVICES

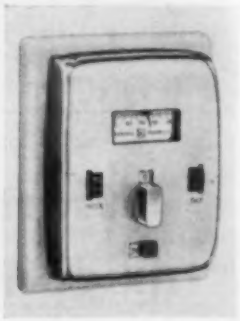
Glycol Vaporizer Especially Designed for Public Buildings

No. 4196**—The "Germatrol" glycol vaporizing unit is custom-built to individual specifications and is also available in three standard sizes. The manufacturer is Germatrol Corp., 622 Brookline Blvd., Pittsburgh.

According to the company, it may be easily connected to the ducts of an air conditioning or ventilating system and rids the air of most germs, including pneumococcus, streptococcus, bacillus, mirococcus, etc. The unit is especially designed for use in public buildings such as churches, schools, theaters, offices, hospitals, and stores.



Glycol Vaporizer



Thermostat

Day-Night Thermostat Controls Unit Heaters, Similar Equipment

No. 4197—The T-70 timer thermostat is a day-night type unit developed by General Controls Co., 801 Allen Ave., Glendale 1, Calif., for controlling space heaters, unit heaters, etc.

Of the two wire, bimetallic type, it has snap action contacts and drum dials. Temperatures between 50 and 90 F may be selected for the night period and the clock mechanism automatically returns the heating system to the preselected day temperature each morning. The night setting period (13½ hr maximum) is shown on a direct reading dial. The daytime setting may be prematurely resumed at will by manually turning the timer to zero.

Two models are available; one providing external day and night settings, the other with external day setting and internally adjustable night setting.

Turbine Type Pump Has Ball Bearings

No. 4198—A turbine type pump featuring an impeller fastened to an enlarged shaft by a lock nut and a woodruff key and sealed in grease ball bearings has been developed by Roy E. Roth Co., Rock Island, Ill.

The bearings are said to eliminate the dangers of bearing rust or improper lubrication. Bearing end play, says the firm, is eliminated by the use of a preloading spring under the outer adjusting nut. This spring is adjusted to the proper tension for maximum bearing life. Elimination of end play is said to insure absolute location of the shaft and impeller so as to avoid metal-to-metal contact. Properly adjusted, the impeller does not touch either side plate.

Other features include elimination of "freezing" and stalling due to temperature changes when handling hot liquids, the development of high pressures in single stage construction, and the use of heads and impellers that are renewable when the capacity of the pump is to be changed.

W. J. Killian OF HERMAN NELSON
William J. Killian is a veteran of the Armed Forces, of the heating and ventilating industry and of The Herman Nelson Corporation. During his 10 years with Herman Nelson and 18 years in the industry, he has gained invaluable experience in solving heating and ventilating problems in industrial, commercial and institutional buildings. Mr. Killian is now Branch Manager of the Cincinnati territory which includes sections of Ohio, Indiana and Kentucky. He is a member of the American Society of Heating and Ventilating Engineers. World War II found "Bill" in the U. S. Navy, where, at the time of his discharge and after active service in the Pacific Theater, he attained the rank of Lieutenant Commander.
NOW YOU FIT INTO THE PICTURE

Seventy-five key men — including WILLIAM J. KILLIAN — work industriously, day in and day out, to provide you with clear, concise and practical solutions to your heating and ventilating problems. These men are Herman Nelson Branch Managers and Product Application Engineers. They are a carefully selected, thoroughly experienced group of men who know all phases of heating and ventilation of industrial, commercial and institutional buildings.

Strategically spotted from coast to coast, Herman Nelson Product Application Engineers specialize in studying the "dollar and cents" angle of your problems as well as technicalities involved. They regard your customers as their responsibilities . . . and their work is finished only when there is complete assurance that Herman Nelson Products are functioning to the customer's entire satisfaction. Herman Nelson service, however, extends beyond these capable representatives. More than 200 Distributors and Stocking Jobbers, with personnel trained in the application and installation of Herman Nelson Products, work closely with Herman Nelson Branch Offices and Product Application Engineers.

There is no substitute for prompt, efficient service — which is probably one of the principal reasons why Herman Nelson has been a leader in the heating and ventilating field for nearly half a century.

herman nelson moline illinois

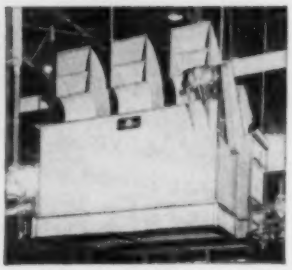
THE HERMAN NELSON CORPORATION

Since 1906 Manufacturers of Quality Heating and Ventilating Products

MOLINE, ILLINOIS

New HIGH PRESSURE STEAM Fan Heater Gives Savings You Never Had Before

The first heater to make High Pressure Steam really trouble-free and practical in plant heating. Every engineer should understand its original method; write for Niagara Bulletin and performance data.



HOW IT OPERATES: —

Dual coil system uses both sensible and latent heat. High pressure steam enters upper coil, shown on diagram below. Condensate drains into a trap. Then this high pressure condensate is released into the header of lower coil, instantly flashing into steam at vapor pressure.

Any high pressure condensate that remains liquid is carried to vapor condensate return header by drain tube that also gives off its heat into air stream.

Vapor condenses in lower coil. Its condensate is kept at a high level in return leg by a wier in the return header and sub-cooled by contact with coldest air entering heater. Vacuum condensate return to boiler is vapor free.

HOW IT SAVES IN COST: —

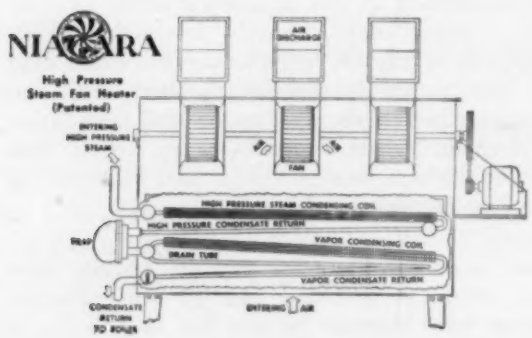
Piping is much smaller, simpler and less costly than in low pressure systems. Big traps and valves are not needed.

HOW IT SAVES IN OPERATION: —

Waste is prevented. Every BTU goes where you want it. No dump traps or hot wells waste live steam.

HOW IT SAVES IN UPKEEP: —

Condensate flow is even, vapor free, easily handled. No sudden surges of condensate in starting. No hammering, no hard wear and tear on system. Final air temperatures are not excessive; heat easily directed where needed; no flashing of low pressure condensate.



NIAGARA BLOWER COMPANY
Dept. H.P.
405 LEXINGTON AVE., NEW YORK 17, N. Y.

Please send Bulletin 109 on the Niagara High Pressure Steam Fan Heater.

Name .....

Address .....

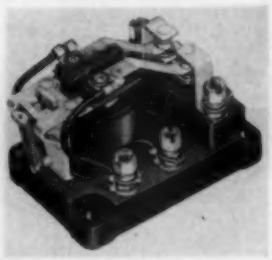
Relays Designed for Control Use

No. 4199—Ward Leonard Electric Co., Mount Vernon, N. Y., has developed Bulletin 110 multi-pole, midget magnetic relays designed for general and special purpose control use. Typical applications include heater, traffic signal, machine tool, alarm, and similar controls.

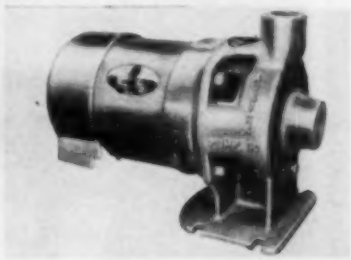
Features are said to include long mechanical life, compact sturdy construction, self cleaning and self aligning contacts, good vibration resistance characteristics, and corrosion protected metal parts.

Standard two or three pole double throw relays have molded phenolic insulating bases, screw terminals for front connection, and silver-to-silver contacts. Leads from movable leaf spring contact fingers to terminals are insulated with glass fiber tubing.

Coils are available for operation on all standard voltages and frequencies up to 115 volts, a-c or d-c.



Control Relay



Small Pump

Small Capacity, Side Suction Pumps Are Close Coupled

No. 4200—Two new small capacity, side suction, close coupled pumps are offered by Aurora Pump Co., 80 Loucks St., Aurora, Ill., for general service and as an integral part of air conditioning units, cooling towers, evaporative coolers, etc.

The type JMC unit, shown here, is designed for capacities of 5 to 75 gpm at heads from 0 to 100 ft and for operating speeds of 1750 to 3500 rpm. It is also available for v belt drive or with a flexible coupling (pump and motor on a common base). Sizes are 1, 1¼, and 1½ in. and units may be obtained with bronze fitted, all iron, or all bronze construction. The casing is vertically split and the bronze impeller is of the enclosed type and is balanced.

The type SAC pump is designed for capacities from 5 to 85 gpm at heads from 0 to 45 ft. Available sizes are ¾, 1¼, and 2 in. Suction is at the end of the unit and the discharge can be rotated to any of several positions 60 deg apart.

Hydraulic Drive Protects Motors From Shock, Vibration

No. 4201**—The new "Hydro-Sheave" drive, manufactured by Twin Disc Clutch Co., Hydraulic Div., Rockford, Ill., is specially designed to be used with Worthington Pump and Machinery Corp.'s quick detachable sheaves made for SB and SD hubs.

The drive is applicable to electric motors and internal combustion engines in the ¾ to 25 hp range. Even acceleration and sustained smoothness of operation are said to result from the application of one of these hydraulic units which eliminates the solid connection between the power source and the driven equipment.

WHEATLAND 49
a link
in a mighty
chain . . .
Wheatland steel pipe used in American industry is but one link in the mighty chain of production and distribution that supplies American homes with the manifold products of thousands of manufacturers. Nonetheless, upon its performance depends the successful outcome of great industrial enterprise. We at Wheatland recognize and accept our responsibility to maintain this vital link strong by supplying steel pipe of the utmost quality.
-pipe with the yearmark!
WHEATLAND-49
WHEATLAND TUBE COMPANY
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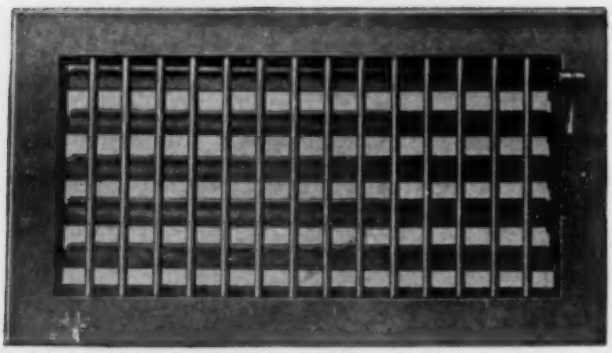
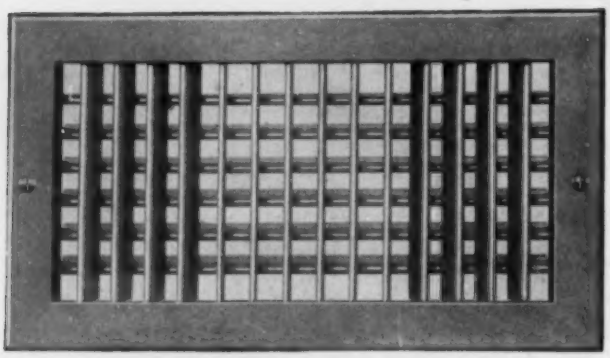
New AUER de luxe

"Streamliner"

Multiple Deflection
Air Conditioning
REGISTERS and GRILLES
— for high velocity outlets



1005V-HML. Single bank of bars, multi-louvre valve.



1205VH. Double bank of bars, no valve.

Streamliner registers and grilles for air conditioning systems provide *better* diffusion for high velocity outlets. The Series includes models with single bank of adjustable bars (vertical or horizontal), also with double bank of adjustable bars (vertical in front and horizontal in back, or the reverse), also all above four types with horizontal multi-louvre valves in the rear, controlled by lever on face of register.

Design of bars and method of mounting them are *original* with Auer. Bars are hollow moulded, beveled, perfectly formed, smooth in contour, and pivoted in the frame on special split sleeves. This expansion insert furnishes correct tension to hold bars firmly where set, yet allows sufficient free movement for adjustment with one simple tool. Bars may be set at any angle, in any grouping. Bars are slightly "tear-drop" shaped, like an airplane wing, and can be turned with thin edge either inward or outward. Blades of multi-louvre valve are pivoted on rivets to prevent vibration. Streamliner registers are equipped with gaskets. You will be pleased with the beauty and fine craftsmanship of these registers. Write for folder S-49.

Complete Auer Register Book—or Catalog "G" on perforated grilles—sent on request.

THE AUER REGISTER CO., 3608 Payne Ave, Cleveland 14, Ohio

Auer REGISTERS
& GRILLES for AIR CONDITIONING & GRAVITY

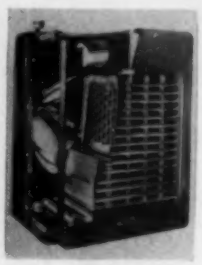
When combined with a QD sheave, the drive is an integrated power transmission package which is designed to protect both the power source and the driven equipment from shock loads and torsional vibration. The sheave is located between the coupling and the power source to eliminate heavy overhanging weight loads.

Cartridge Type Ribbon Burner Features Gas Fired Unit Heater

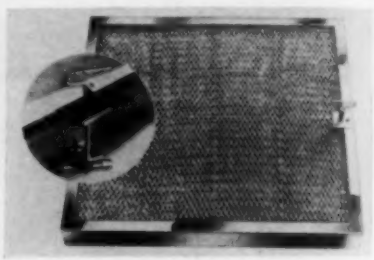
No. 4202—A completely redesigned line of "Janitrol" unit heaters, UBS series, is now being produced by Surface Combustion Corp., Toledo 1, Ohio.

A new cartridge type ribbon burner is said to provide markedly increased combustion efficiency and to weigh only 30 percent as much as the standard type of cast iron burner. Improved "Multi-Thermex" steel tube heating assembly units are now incorporated in these heaters. According to the company, the steel tubes of the heat exchanger are produced by a new type of cold rolling mill which can roll corrugations in steel sheets without stretching or thinning the metal at any point in the corrugations.

The new heaters are approved by both the AGA and the Underwriters' Laboratories. Sizes range from 50,000 to 225,000 Btu per hr input.



Unit Heater



Grease Filter

Handle Lock Used to Fasten Air, Grease Filters

No. 4203—A new handle lock designed for installation on R-P "Alumaloy" grease and industrial air filters is available from Research Products Corp., Madison 10, Wis. It may be used with any standard size of these two filters.

The new handle lock is said to provide rattle-free fastening that compensates for possible variation in frames or loosening due to wear. This is accomplished by means of a helical spring which compensates for varying holding frame flange dimensions, or for flattening of felts, and similar factors. The center shaft is so designed that it can be adjusted for additional compensation after use. Once assembled, a quarter turn of the handle in either direction moves the latch rod over a built-in cam which lifts the rod to clear any commonly encountered holding frame flange.

According to the company, most of these air and grease filters already have provisions for quickly mounting the new handle locks.

Motor Designed for Deep Well Turbine Pumps

No. 4204—The type CSP "Life-Line" motors, especially designed for deep well turbine pumps by Westinghouse Electric Corp., P. O. Box 868, Pittsburgh 30, are vertical, hollow shaft units of weatherproof construction.

Newest Instrument

for fast, accurate reading of low surface temperatures

Insulating Contractors

Heating Contractors

Air-Conditioning and Refrigeration Contractors

Here's an instrument that answers a real need for those interested in easy, accurate measurement of low surface temperatures . . . up to 150°F.

The new Alnor Type 500 Pyrometer is a highly portable, self-contained direct reading instrument that accurately indicates surface temperatures . . . fast! A full reading requires only 3 to 5 seconds! These features make the Type 500 an ideal instrument for architects, heating and air-conditioning contractors, refrigerator contractors, insulating contractors, and all whose work involves checking relatively low surface temperatures accurately . . . and easily.

If low temperature measurement is important to you, use the coupon to get your copy of bulletin 924. You'll find the Alnor Type 500 Pyrometer ideal for speed and accuracy . . . the price surprisingly low.

The Alnor Type 500 Pyrometer for measurement of relatively low surface temperatures . . . from —50°F. up to 150°F.

Alnor

PRECISION INSTRUMENTS FOR EVERY INDUSTRY

ILLINOIS TESTING LABORATORIES, INC.

Room 504, 420 N. La Salle Street, Chicago 10, Illinois

Illinois Testing
Laboratories, Inc.
Room 513, 420 N. La Salle Street
Chicago 10, Illinois

( ) Send me bulletin 924.
( ) Please have an Alnor representative call.

Name .....

Address .....

City .....Zone.....State.....

Less labor to install . . . .
more heat at less cost!
There is no simpler or more effective heating equipment than
KRITZER
COMMERCIAL
"FIN-PIPE" COILS
for Walls, Overhead or Floor-Level Heating

Kritzer Radiant Coils have been designed, engineered, and produced, to give you efficient, low cost, fast heat transfer with a simple, rapid, labor and dollar-saving installation.

Fins grip pipe with specially designed deep collars to assure you better heat transfer and a permanent, positive bond. Kritzer industrial coil sections are furnished with chamfered ends ready for welding, or threaded with modern, improved chaser type die heads that give clean, sharp threads so necessary for fast, leak-proof connections.

IF IT'S KRITZER IT'S RIGHT SIR

Exclusive with Kritzer is the new notched fin design. Grilles of all types snap on to coil quickly, in perfect alignment. They may be removed—*but they'll never fall off!*

This coupon will bring you factual information about "FIN-PIPE" heating for commercial use . . . for stores, factories, offices, public buildings, schools, institutions, etc. Write today.

KRITZER RADIANT COILS, INC.
2905 W. Lawrence Ave., Chicago 25, Ill.

Gentlemen: I'm interested. Please send complete information and bulletin on Kritzer Commercial "Fin-Pipe" Coils.

Name .....

Firm .....

Address .....

City ..... Zone ..... State .....

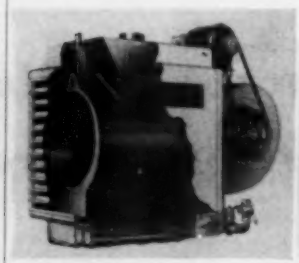
Of the squirrel cage induction type, they can be furnished as a complete motor, a motor without base, or as a stator and rotor only. They are offered in sizes from 1 to 20 hp for 208, 220, 440, and 550 volts, 60, 50, or 25 cycles.

Typical uses are said to include paper mills, irrigation work, construction projects, municipal pumping stations, etc.

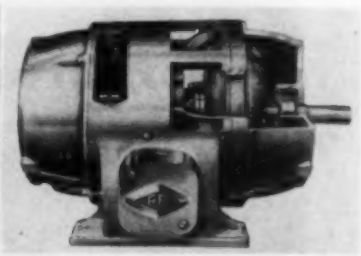
Gas Unit Heater Has Centrifugal Blower

No. 4205—A line of new "Humphrey" rotor type, gas fired unit heaters is available from General Gas Light Co., Kalamazoo, Mich.

The chief feature is the use of a centrifugal fan designed to operate against duct resistance and to deliver heated air to remote areas. Adjustable louvers direct heated air downward to working zones when ductwork is not used. Units are completely automatic and are offered in five sizes from 65,000 to 200,000 Btu per hr input. Air deliveries at ¼ in. SP range from 950 to 2110 cfm.



Unit Heater



Motor and Fluid Drive

Motor and Fluid Drive Designed as Integral Unit

No. 4206**—The "Fluid-Shaft" motors of Reuland Electric Co., Alhambra, Calif., feature a single frame and integral design of motor and fluid drive coupling. This, together with utilization of the firm's electric motor frames and end bells, is said to result in noteworthy savings in both original cost and mounting area.

Units are recommended for use wherever loads require smooth acceleration, protection from "jamming" or shocks, or are difficult to start. Other advantages are said to include the selection of motors close to actual horsepower requirements and a definite saving in power because of a considerable saving in starting current.

Available in standard foot mounted or round body frames, all units can be mounted horizontally or vertically, or with NEMA flange and face type end bells. Sizes are ½, ¾, 1, 1½, 2, 3, 5, 7½, and 10 hp. Also available are speed reducing units of either the single or double reduction type.

Exhaust Fans Designed for Vertical or Horizontal Discharge of Air

No. 4207—Six new ball bearing, belt driven exhaust fans in 36, 42, and 48 in. sizes are being marketed by Emerson Electric Mfg. Co., St. Louis 21.

Equipped with ball bearing motors and ball bearing pillow blocks to absorb the thrust load, they are said to be suitable for air discharge at any angle. Features include capacitor motors, rubber hub mounting, thermal overload protection, and four bladed propeller type fans designed for large volume delivery and quiet operation. Capacities are from 10,200 to 19,350 cfm.

1 plan helps prevent all 3

When you show the people in your company that you're interested in their welfare, they naturally react favorably. That's why *production goes up* when you boost your employee-participation in the Payroll Savings Plan. Here's how it happens:

The more U. S. Savings Bonds an employee holds, the more secure he feels. The more secure he feels, the greater his peace of mind—the more contented, careful, and productive he is on the job.

Those facts aren't just something we dreamed up. They're borne out in the experience of more than 20,000 companies promoting the Plan.

Everybody Benefits!

Bond sales spread the national debt, thus increasing our national economic security. And, of course, what's good for the nation is good for you and your business!

The individual Bond buyer gets back $4, when his Bonds mature, for every $3 he invested. That's a boon for him, and —multiplied by millions of Bond holders —represents a huge backlog of purchasing power that will help assure national prosperity through the years ahead.

Five Steps Boost Participation

1. See that a top management man sponsors the Plan.

2. Secure the help of the employee organizations in promoting it.

3. Adequately use posters and leaflets and run stories and editorials in company publications to inform employees of the Payroll Savings Plan's benefits to them.

4. Make a person-to person canvass, once a year, to sign up participants.

These first four steps should win you 40-60% participation. Normal employee turnover necessitates one more step:

5. Urge each new employee, at the time he is hired, to sign up.

Nation-wide experience indicates that 50% of your employees can be persuaded to join—without high-pressure selling. All the help you need is available from your State Director, U. S. Treasury Department, Savings Bonds Division. He is listed in your telephone directory. Wouldn't it be a good idea to call him right now, while it's on your mind?

The Treasury Department acknowledges with appreciation the publication of this message by

Heating Piping & Air Conditioning

This is an official U. S. Treasury advertisement prepared under the auspices of the Treasury Department and The Advertising Council.

JOHN ZINK CONVERSION BURNERS
Available In Two Models

Luminous Flame Model Illustrated

LUMINOUS FLAME MODEL
Easy to install. Operates manually or automatically. Burns clean fuel—natural, mixed, manufactured or LP gases — without soot or smoke. Luminous flame radiates heat two to three times faster than a blue flame. Four sizes available for either vertical or horizontal firing.

BLUE FLAME MODEL
Designed to meet the requirements of this type of burner. Capacity range of 100,000 to 200,000 Btu/hr. for operation on natural, mixed or manufactured gases. Manual or automatic operation. Easy to install.

JOHN ZINK UNIT HEATERS

Whisper Quiet Model

WHISPER QUIET MODEL
For installations requiring quiet operation. More Head Room—Compact Design and More Heat From Gas Burned. A. G. A. approved for natural, manufactured, mixed or LP gases. Fully equipped with Automatic Controls.

HURRICANE MODEL
For Industrial Installations. This model is also compact in design, offering more Head Room and More Heat from Gas Burned. Will heat 40' x 80' space. A. G. A. approved for natural, manufactured, mixed or LP gases. Automatic Controls.

100,000 Btu/hr.

Both Models finished in Toast Tan Crinkle Enamel Baked on. Adjustable Louvers.

Write for Literature

John Zink Company

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TULSA, OKLAHOMA

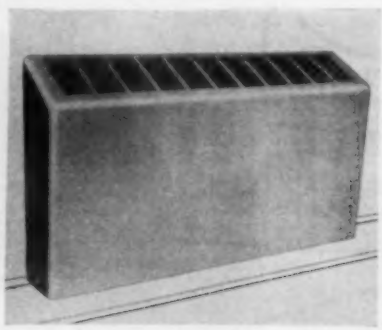
New York — Salt Lake City — Houston — Los Angeles

Adds Wall Hung Type to Line of Convectors

No. 4208—Tuttle & Bailey, Inc., New Britain, Conn., has added wall hung copper convectors to its line of standardized convector types.

Designed for installation in schools, hospitals, institutions, offices, and dormitories, where frequent cleaning of the floor area is essential, units have slanted tops (30 deg) to prevent their being used as "tables" or "ladders." Tops and fronts are furnished in one piece and are constructed of 18 ga steel, bodies are of 20 ga. For applications where extra strength is needed, heavier gages are available.

The new convectors are furnished with all-copper heating elements and chain operated dampers can be attached where required.



Wall Hung Convector

Electrostatic Cleaner

Electrostatic Air Cleaner Developed for Use With Packaged Conditioners

No. 4209—Trion, Inc., has developed an electric air filter especially for use with 3 and 5 ton packaged air conditioners.

The model 904 unit is designed to handle air volumes up to 2000 cfm and to be 90 percent efficient in removing dust, dirt, smoke, soot, pollen, and airborne bacteria. It is said to require very little floor space and is recommended for use in offices, shops, food stores, restaurants, laboratories, photo supply houses, radio and telephone control rooms, and other applications where clean air is essential to greater profit and comfort.

The filter unit is shown here with a Chrysler Airtemp packaged conditioner.

Small Check Valves Made Possible by New Body Design

No. 4210—A new cast steel valve body, developed by Edward Valves, Inc., East Chicago, Ind., is said to make possible small, economical check valve installations because of unusually low pressure loss.

Smaller body size has been achieved by the use of an integral stellite seat and the company's equalizer which allows the disk piston to rise completely out of the path of flow, thus utilizing the entire flow passage. The bonnet bore is said to be only slightly larger than the ASA standard for port openings. As a result, overall weight of the valve is reduced and pressure load on the bonnet seal and operating mechanism is minimized. The tubular design of the disk piston is used to insure minimum weight and to prevent over-the-disk flow.

Check valves with new body design are built in 900, 1500, and 2500 lb classes with both globe and angle patterns. They are available with pressure seal or bolted bonnet and with flanged or welding ends.

Air Velocity Meter Combines Hot Wire, Thermopile Principles

No. 4211—Hastings Instrument Co., Inc., Box 1275, Hampton, Va., has announced an air meter incorporating two ranges for measurement of air velocities in the range of 0 to 400 and 400 to 6000 fpm.

Designed to measure air velocities as low as 5 fpm, it is applicable to the measurement of air velocities in ducts, flow in air and gas lines, surveys of heating and air conditioning systems, process control and meteorological studies. Operation of the instrument is based on a combination of the hot wire and thermopile principles. This arrangement, says the company, increases accuracy by eliminating errors due to air temperature variations. The output is said to be suitable for remote indication and for operating a standard strip or circular chart electrical recording instrument. The device operates from 110 volts a-c power.

Air Velocity Meter

Arc Welder

Extra Starting Current Aids Arc Welder Operation

No. 4212—A record low price on its new 200 amp a-c industrial type welder has been announced by Lincoln Electric Co., Cleveland. The "Fleetwelder 200" incorporates an "Arc Booster" and is said to have outstanding reserve capacity, making the unit ideally suited for both job shop welding and industrial welding.

It is designed for a wide range of work in thick as well as thin material. Electrodes ranging in diameter size from 5/64 to ¼ in. may be used with it. The booster is designed to adjust the welder to start the arc automatically the instant the electrode touches the work on either thin or heavy material. An extra burst of current is used to start the arc and secure penetration at the start of the weld. The current automatically returns after a few seconds to the correct amount set for the job.

Condensation Return Units Available With Steel or Cast Iron Receivers

No. 4213—Series UC-8 (steel receiver) and US-8 (cast iron receiver) condensation return units consist of a 15 gallon receiver, a Worthington "Monobloc" centrifugal pump, a splashproof motor, and the necessary interconnecting piping, float control, and wiring for automatic operation. The manufacturer is Faber Engineering Co., Inc., Amber and Westmoreland Sts., Philadelphia 34.

Units are available with capacities from 1000 to 8000 sq ft EDR and for discharge pressures from 15 to 40 psi. These compact, low cost units are said to be dependable, quiet, and economical in operation. They are equipped for 115 volt operation but may be connected to 230 volts by the purchaser's electrician.

SIMPLE A-B-C's FOR REINSULATION OF UNDERGROUND PIPING

A—Remove roof slab. Clean out old insulation and metal supports. Repair box.

B—Apply parting medium to pipes and fill box with Z-Crete insulating concrete.

C—Reinsulation is completed and is ready for backfilling.

"Z-crete"* Insulation

is a special mix of lightweight cellular concrete, formulated exclusively for use on underground heated piping. It is a permanent type of insulation which does not rot or disintegrate if wetted and so is ideal for reinsulation of box conduits. It is efficient because it is applied thicker and the solid nature of the fill eliminates the possibility of convected air losses. It is dielectric and provides an excellent corrosion protection for the embedded pipes.

Licensed applicators in all principal cities.

*Z-Crete is a registered trademark of Zonolite Company

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Right here, in these two buildings, you'll find the very latest products and ideas for heating, ventilating and conditioning air in every type of building. More than 200 exhibitors and their technical staffs will be on hand to discuss and demonstrate the most economical, efficient and durable installations for all purposes.

No industrial or consulting engineer, manufacturer, contractor, distributor, building owner or operator can afford to miss this chance to get new business-building ideas, make new contacts, see new products.

So make plans now to attend this most instructive, valuable exposition . . . and bring your associates.

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Packless Expansion Joint Has Monel Metal Bellows

No. 4214—The "Sylphon" packless expansion joint of Fulton Sylphon Div., Robertshaw-Fulton Controls Co., Knoxville 4, Tenn., is now available with a monel metal bellows.

According to the company, this improved model assures longer life, higher pressure limits, a vacuum tight piping system, and greater resistance to corrosion and erosion. It is particularly recommended for absorbing heating riser expansion in the taller buildings.

Screw Fasteners Made in Continuous Strips

No. 4215—Flat-type "Speed Nuts," now being produced in continuous strips, permit savings in assembly time, according to Tinnerman Products, Inc., 2035 Fulton Rd., Cleveland.

Called the tandem type nut, the new product consists of a strip of pretempered spring steel into which the fasteners are stamped, a partial shear separating each unit. This permits the operator to break the strip from the leading nut as it is tightened.

According to the company, the leading nut is at the tip of the screw driver at all times, there is no loss from dropping on the floor since the nuts are fastened together, and locating the nut and starting the screw is facilitated since the longer strip is handled easier than an individual nut.

The new nuts are produced in either strips or coils. The fastener itself is a curved strip of spring steel with two arched prongs formed to engage the screw threads and designed to lock permanently.

Equipment Shorts

American Blower Corp., Detroit, has completed negotiations for a new manufacturing and assembly plant in San Leandro, Calif. Plans call for the erection of a single building covering 40,000 sq ft. ..... All manufacturing facilities of **Baker Refrigeration Corp.** will be concentrated at its eastern division in South Windham, Me. Negotiations have been completed for approximately 35,000 sq ft of additional manufacturing floor space. The company plant in Omaha, Nebr., will be closed, but the firm's branch in that city will be expanded as a service and parts depot.

In the annual report survey of Financial World, **The Trane Co.**, La Crosse, Wis., was judged as having the best 1948 annual report of the building equipment industry. The report of **Iron Fireman Mfg. Co.**, Cleveland, was judged best in the stove and heater industry. Bronze trophies were presented last month at the award banquet. ..... **Niagara Blower Co.**, New York, has been

awarded a license to manufacture dehumidifying and air conditioning equipment under a series of patents of **Research Corp.**, New York, which cover the removal of moisture from air or other gases by the use of a hygroscopic liquid.

Jas. P. Marsh Corp., Skokie, Ill., has purchased the **Electrimatic valve division of Simoniz Co.**, Chicago. Sales of the newly acquired line of refrigeration control valves and regulators are to be handled by Electrimatic Co. Sales of Marsh pressure gages, dial thermometers, and related products will be handled by Marsh Instrument Co. As in the past, heating specialties will be handled by Marsh Heating Equipment Co. Each of these organizations is a sales affiliate of Jas. P. Marsh Corp. The Electrimatic refrigeration fittings, strainers, and dryers of Simoniz have been taken over by the newly formed **Madden Brass Products Co.**, Chicago. Both M. B. Madden, who heads the new firm, and F. H. Moss, sales manager, were with Simoniz.

E. I. du Pont de Nemours and Co., Inc., Wilmington, Del., has disclosed increasing production of the newly developed metal, titanium. The first small scale production of the metal was announced by the company a year ago. Plant capacity was about 100 lb per day. This has been increased and plans are under way to increase production further next year. Future plans depend upon the uses developed for it by the metals industry. According to the company, the metal is silver white, light, and strong, with excellent corrosion resistance. It is said to be comparable to stainless steel in strength but to weigh little more than half as much.

Elliott Co., Jeannette, Pa., has acquired the business and assets of **Crocker-Wheeler Electric Mfg. Co., Div. Joshua Hendy Iron Works**, Ampere, N. J. Formerly the Elliott line did not include motors under 200 hp, but will now consist of a-c and d-c motors from 1 to 25,000 hp. The Crocker-Wheeler Div., Elliott Co. will continue to operate under its previous management. ..... A new, modern manufacturing plant has been opened in Houston, Tex., by **Link-Belt Co.**, Chicago. The building includes a two story office section and a one story factory section consisting of a machine shop, structural steel shop, and warehouse facilities.

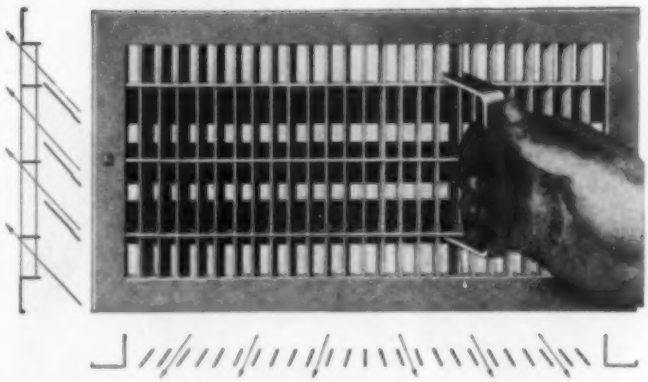
Marlo Coil Co., St. Louis, has reduced prices approximately 20 percent on evaporative condensers for ammonia and industrial cooling units for ammonia. The reduction is said to be in line with recent reductions in the price of hot galvanizing. ..... Incorporating its trade name, **Powermatic Ventilator Co.**, Cleveland, has changed the firm name to **Iron Lung Ventilator Co.** The headquarters office remains at the same location.

Last July, **Union Malleable Mfg. Co.**, Ashland, Ohio, completed a modernization program. A further expansion, recently announced, will be an addition to its plant in Ashland. As part of a new packaging program, the firm will ship products in packages for ease of sales and storage. Bulk shipments will continue to be made. A new "U-Cote" process has been developed by the firm and is being used to protect its black malleable iron fittings.

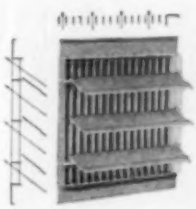
INDEPENDENT

"Fabrikated"

WALL GRILLES WITH DEFLECTING VANES



Style 321-A grille with deflecting vanes



Rear view showing adjustable deflecting vanes

ACCURATE and positive compound deflection of air flows can be readily secured with these "Fabrikated" grilles. Directional adjustments may be made when grilles are installed; or grille bars and vanes may be adjusted after installation to meet unforeseen or changed requirements.

Each grille bar and each deflecting vane is adjusted *individually* with a special two-pronged tool included with each shipment. Locking of grille bars and vanes is not required because they are held firmly in place — no vibration — no rattle.

Write for Our New Catalog No. 48.

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Always Leading—Always Progressing

THE INDEPENDENT REGISTER CO.

3747 E. 93rd STREET · CLEVELAND, OHIO

SOLVE DIFFICULT
STACK PROBLEMS
this
ECONOMICAL WAY

with NEW
DeBOTHEZAT
INDUCED DRAFT
BIFURCATOR
FAN

- OFTEN ELIMINATES COSTLY STACK CONSTRUCTION
- PERMITS CONTROLLED DRAFT REGARDLESS OF WEATHER
- SAVES SPACE AND OPERATES MORE EFFICIENTLY than CENTRIFUGAL FAN

DeBothezat Induced Draft Bifurcator Fans can be installed in vertical stack or horizontal or angular breeching. The Bifurcator is a direct motor driven fan with a divided housing. Gases by-pass motor which is isolated in through-ventilated, asbestos lined chamber. For operating temperatures up to 650° F. Also special construction for higher temperatures.

Fan can be connected to firing controls, for uniform draft. Motor easily accessible. No belt, countershafts or platforms required.

FAN SIZES 12" THROUGH 27". For Boilers from 6 HP (200,000 BTU per hour) up to 350 HP (12,000,000 BTU per hour).

Write today for complete details and prices.

DeBOTHEZAT FANS DIVISION
AMERICAN MACHINE AND METALS, INC.
Dept. H-11, East Moline, Illinois

Thousands of small, 3 in. blowers, made by **Sturtevant Div., Westinghouse Electric Corp.,** Hyde Park, Boston, are being used in the Navy's "mothball" fleet as part of dehumidifying systems installed aboard inactivated ships. The fan in each unit circulates air in the sealed off compartments through silica gel and is also used for reactivation.

A new and simpler process for welding boiler tubes has been developed by **Combustion Engineering-Superheater, Inc.,** New York. Said to make possible a consistently high quality of weld, the process employs the principles of induction heating and pressure welding. A feature is the use of an enclosed atmosphere to protect the welding surfaces against oxidation. Both bent and straight tubing may be welded by this method without upsetting of metal in the welding zone.

Wagner Electric Corp., 6328 Plymouth Ave., St. Louis 14, has developed a new hydraulic dynamometer, SA-484, of 30 to 300 ft lb capacity for load testing large electric motors. Adaptors for shaft sizes from 1⅛ to 2⅞ in. are furnished. Tests that can be made include idle, pull-in, pull-up, full load, and locked. The unit cannot be used to make a heat run on a motor. . . . "Chromasoid" is a new low cost lining material for corrosion prevention on all condensate tanks and return system receivers of **Fred H. Schaub Engineering Co., Inc.,** 2110 S. Marshall Blvd., Chicago 23. According to the company, the material is not affected by any chemical reagent, will withstand severe thermal shock, does not impart color, odor or taste to water, and is unaffected by temperatures up to 500 F.

Celotex Corp., 120 S. La Salle St., Chicago 3, has added a new batt type rock wool blanket to its line of insulating materials. Improvements are said to include a more efficient vapor barrier and an improved rock wool filler that is better felted, lighter in weight, and more resilient. Blankets are 15 x 24 in. and 15 x 48 in. . . . Developed during the war for Navy fire fighting teams, "Chemox" is a new type gas mask approved by the U. S. Bureau of Mines. It is available from **Mine Safety Appliances Co.,** 201 N. Braddock Ave., Pittsburgh 8, and is said to be the only respiratory protective equipment that generates its own oxygen supply. The self contained apparatus weighs 13½ lb and is designed to permit remaining 45 min in an area that has an oxygen deficient atmosphere.

United States Testing Co., Inc., Hoboken, N. J., has expanded its services to include the qualification of welders for welding contractors and steel fabricators. The new service permits an employer to send a prospective welder employee to the company where qualification tests are made and weldments tested.

Randall Graphite Bearings, Inc., Chicago, has purchased the foundry and machine shop of **Shook Bronze Corp.,** Lima, Ohio. The foundry was modernized in 1936 and the machine shop will be augmented by a transfer of the firm's Chicago plant to the new acquisition, to be known as Randall Graphite Bearings, Inc., Shook Bronze Division. Alan Dale, formerly vice president and sales manager for Shook, is now assistant to the Randall president.

To speed up the analysis of boiler scales, **Hall Laboratories, Inc.**, Pittsburgh, is burning samples in a d-c arc where the intense heat breaks down practically all compounds into their elements. Most of these are gases at such temperatures and give off light of characteristic wave lengths which are "sorted out" by a spectrograph. This method, while relatively quick, does not identify a compound as such, but rather some of the constituent elements. For this reason it is said to complement, rather than replace, other laboratory tools and techniques.

Met Sales & Mfg. Co., 1217-C N. Main St., Rockford, Ill., has developed a "Handi-Heat" tinner's furnace burning bottled gas. Containing a burner, furnace, and fuel tank, it is recommended for sheet metal, construction, pipe fitting, and electrical applications. . . . The portable incinerators of **Plibrico Jointless Firebrick Co.**, 1800 Kingsbury St., Chicago 14, may now be equipped with water heating units. Capacities range from 50 to 200 gph at 100 deg rise (with 1200 F furnace temperature over the heater) depending on the size of the incinerator used and whether the heater is the single or double type.

A hydro-pneumatic pressure tank calculator, available from **Automatic Control Equipment Co.**, 1005 University Ave., St. Paul 4, Minn., is a slide rule type instrument for determining the most suitable air-water ratio, the most efficient pressure differential, the amount withdrawn on any pressure drop, and the resultant reserve or residual. . . . Now in production by **Actus Corp.**, Mount Vernon, N. Y., the new "Eagle" hose clamp employs a hinge principle and a gear type screw for take-up after the clamp is in place. It is available in five sizes to cover hose from ⅜ to 3¼ in. ID.

Wall Catalog Co., 59 E. Madison St., Chicago 3, has prepared a number of wall charts showing architectural and engineering symbols. One of them gives symbols for heating, ventilating, and air conditioning equipment and includes a table of decimal equivalents. Charts are available in bulk quantities with company imprint or in single copies at $1.00.

New and Revised Equipment Standards in ACRMA Supplement

Four new recommended standards and three revised standards make up the new 1949 Supplement to the Equipment Standards of the Air Conditioning and Refrigerating Machinery Association.

Published as helpful guides to both manufacturers and purchasers of air conditioning and refrigeration equipment, the standards have been prepared as representing "desirable industry practice." Equipment covered includes room air conditioners, self contained air conditioners, open type refrigeration condensing units for commercial application (air and water cooled, 20 hp and under), high speed ammonia compressors, application of refrigerant heat transfer equipment, water cooling towers for use with refrigerating equipment, and application and installation of compression refrigeration systems.

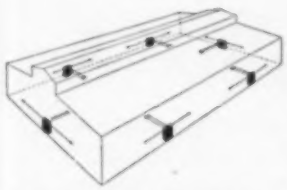
Paper covered copies of the supplement are available for $1.00 from ACRMA headquarters, 717 Southern Bldg., Washington 5, D.C.

Direct Fired

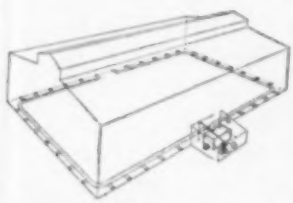
WARM AIR HEATERS*

by LEE

a COMPLETE line for every application



One or more heaters is placed inside of the space to be heated.



Heater is placed in a central location, warm air being delivered by ducts to space heated.

Whatever your needs in large scale heating you will find a Lee Direct Fired Unit to handle the job efficiently and economically. Over thirty years of experience in building quality heaters guarantees the proper unit for the job whether you contemplate a unit or central system. Illustrated at left are the two types mentioned — the top diagram shows an installation using one or more heaters in which capacities from 300,000 to 6 million Btu may be obtained. The bottom diagram illustrates the central system in which the heated air is distributed through ducts in which 3 million to 8 million Btu may be developed.

The efficiency of the Lee counter-current flow reduces fuel consumption; while the rugged simplicity of the system decreases cost of installation, assures low maintenance charges, and also provides ventilation without additional cost. In addition to normal heating installations Lee Direct Fired Units may also be used for drying and curing in cases where these two operations are required.

It will pay you to investigate the Lee line today—literature is available describing completely the principal types of construction and versatility of operation as well as a partial list of prominent and satisfied Lee customers.

Please write us today.

LEE ENGINEERING COMPANY

95 River Street **Hoboken, N. J.**

HERE'S THE NEWEST INSULATION

TYPE 6

INFRA ACCORDION INSULATION

Three thick sheets of permanently separated aluminum

PLEASE LOOK AT THE THERMAL VALUES*

4" to 7½" Insulation Value in a 2½" Depth

There are SIX, 97% effective, radiant-heat-repelling surfaces, EACH of ONLY 3% emissivity (against ordinary insulation's 90%). There are SIX non-conductive, reflective air spaces. FIVE barriers to convection consist of THREE EXTRA-THICK aluminum sheets, IMPERVIOUS to warm or cold air, WATER VAPOR, or any gas; and TWO Fiber Separators, flame, mold and vermin-proof, which also block inner convection. For 16" and 24" centers.

*THERMAL FACTORS, INFRA TYPE 6

Down-Heat	C.044, R 22.72 = 7½"	dry rockwool
Up-Heat	C.08, R 12.50 = 4"	dry rockwool
Wall-Heat	C.073, R 13.69 = 4½"	dry rockwool

Infra — Light, Strong, Easily Installed, Competitive in Price

WRITE Infra for details and FREE COPY of "Bulletin No. 38," issued by the National Housing Agency of the Government, reporting tests of Aluminum Insulation made by the U. S. Bureau of Standards, and dealing principally with the problems of condensation and of heat and vapor transfer. *Address Dept. AC*

Infra

MULTIPLE ACCORDION ALUMINUM & TRIANGULAR REFLECTIVE AIR CELLS

INSULATION, INC.

10 Murray St., N. Y., N. Y.

WHO'S WHAT

Charles E. Walker, Jr., has been elected vice president of **Dravo Corp.,** Pittsburgh. He joined the company in 1933 and for the last five years has been secretary and director of industrial relations, in which positions he continues. ..... **Laurence P. Quinlivan** has been transferred from the Cincinnati branch, where he served as industrial sales engineer, and is now at the headquarters of **York Corp.,** York, Pa., where he is manager of air conditioning sales. He has been with the firm since 1934. **Roger H. Dowling,** who joined the company in 1928 and who was made assistant sales manager in 1943, is the new general service manager.

Buffalo Pumps, Inc., Buffalo, N. Y., has elected **Henry D. Wilson** and **Bruce W. Ellis** to the board of directors. Mr. Wilson continues as factory manager and Mr. Ellis remains chief engineer. ..... **L. W. Clarke,** vice president in charge of sales, **Philip Carey Mfg. Co.,** Cincinnati, is a new director of the company which he joined in 1935 as insulation department manager in the New York office. ..... As a new vice president of **Elliott Co.,** Jeannette, Pa., **Charles A. Butcher** continues as general manager of the Crocker-Wheeler Electric Mfg. Co., Ampere, N. J., a recently acquired division of Elliott.

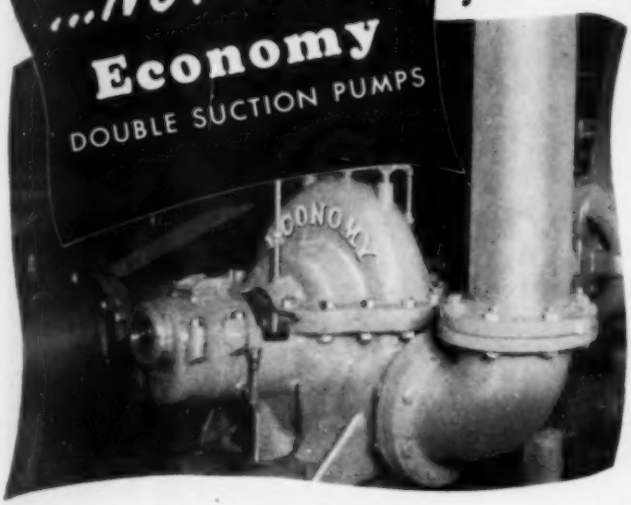
A. C. Traub, formerly manager of dealer sales at the Los Angeles branch of **Fairbanks, Morse & Co.,** Chicago, is now manager of the stoker division and is located at the headquarters office. He has succeeded **F. J. Kluempers** who has joined the administration division of the company. Also located in Chicago is **T. E. Woodruff,** formerly manager of the St. Louis branch house pump department. He is now assistant manager of the pump sales division. ..... **George W. Altman** has been named manager of the Trenton, N. J., cast iron radiator foundry of **National Radiator Co.,** Johnstown, Pa. He joined the company early this year as a foundry engineer, having previously been with Hannifin Corp. as general manager of its subsidiary, Bridgman Castings, Inc.

Thomas M. Cunningham has been appointed to the staff of the **Heating, Piping, and Air Conditioning Contractors' Chicago Association** and will become executive secretary upon the retirement of Louis T. Braun next July. He has been director of industrial relations for Carrier Corp. and is the author of an article on the use of sheet board materials for ventilating and air conditioning duct construction, published in the October 1942 issue of HPAC. ..... **Max L. Murdock,** acting manager, centrifugal pump department, Norwood, Ohio, works, **Allis-Chalmers Mfg. Co.,** Milwaukee, is now manager of the department, succeeding **H. C. Gaton,** retired. **Woodrow Brixius,** formerly application engineer in the company's West Allis works centrifugal pump department, is assistant to Mr. Murdock, and **Paul B. Hugenberg** is now application engineer in charge of sales and orders.

American District Steam Co., North Tonawanda, N. Y., has opened a direct factory sales office at 341 Madison Ave., New York City. Resident manager is **Lawrence D. Toolan** who was formerly eastern merchandising super-

Caught with your pumps down?

...NOT WITH Economy DOUBLE SUCTION PUMPS



12 x 10" Economy Double Suction Pump in white water service in large felt mill. 3000 g.p.m., 40 foot head.

PUMPS that are "down" send production and profits down, too. Keep costly maintenance shutdowns to a new low with Economy Pumps . . . engineered for true economy in operation. Economy Double Suction Pumps, illustrated, are ruggedly built to keep pumping longer . . . with less maintenance. Advanced hydraulic design. Duplex Thrust Bearings mounted back to back for maximum life. Effective cross lubrication to both bearings. Cartridge enclosures protect bearings . . . permitting removal of complete rotor without exposing bearings to dirt and dampness.

For the complete Economy story on Double Suction Pumps, for general water supply or mill service, write Dept. C-11 for bulletin A1147.

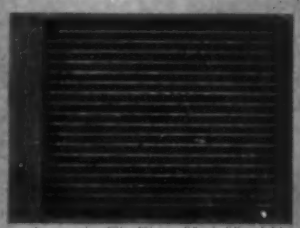
Centrifugal, Axial and Mixed Flow Pumps for all applications.

Economy Pumps, Inc.

Division of Hamilton-Thomas Corp.
Hamilton, Ohio

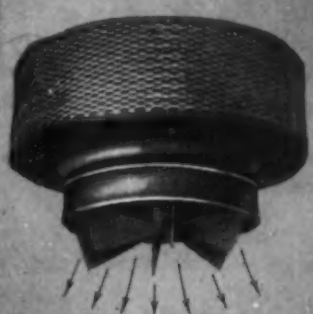
McCORD HEATING AND AIR CONDITIONING PRODUCTS

BLAST HEATING COILS—AIR CONDITIONING COILS—ALL TYPES



All copper spiral fin tube construction with the tubes formed or bent permits uniform expansion without strain. Headers are heavy wall construction with tubes brazed into extended ferrules formed in the headers. Fins are helically wound and metallically bonded. May we send catalog?

VERTICAL TYPE UNIT HEATERS



A proven heating unit, modern in design, incorporating every unit heater advancement. McCord copper spiral fin tube construction creates air turbulence without undue restriction. Improved fan blades deliver large volume of air with minimum air noise. Guaranteed for 150 lbs. saturated steam pressure. Specify McCord.

HORIZONTAL TYPE UNIT HEATERS

A time proven heating unit, modern streamlined in design, quiet, yet its rugged construction adapts it to all types of installations, guaranteed for use on No. 150 p. s. i. steam pressure. All copper heating element, individual spiral fin tubes free to expand without strain on joints. Standard motors with standard bases used. Wide range of capacities.

Save fuel with McCord Unit Heaters.

McCord CORPORATION
DETROIT 11, MICH.

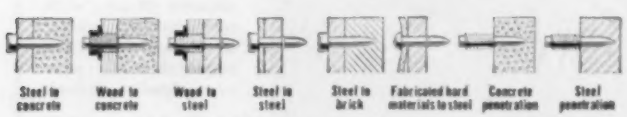
$2,122.00 SAVED

with Ramset FASTENING SYSTEM

A $70 investment in a RAMSET TOOL saved this contractor (name on request) $2,122 *on just one job*. Cost of setting 7,118 fasteners by drilling and tapping would have been $4,742 (plus material). With the RAMSET SYSTEM, the total cost for material and labor was $2,620—saving $2,122 or 44.7%. All for a $70 investment!

SPLIT-SECOND FASTENING WITH RAMSET

No matter how large or how small your fastening jobs, you do them better, quicker, cheaper the RAMSET way. The light, portable tool takes only 30 seconds to load. No electric or air lines—it's self-powered. Press it against the work and RAM—the job is done. So simple and easy that we can teach any good workman, in an hour, to "RAMSET" jobs like these:



LET US ANALYZE YOUR FASTENING COSTS

Ask for 15-minute demonstration of RAMSET SYSTEM. We'll show you how you can do your work faster, better and at amazingly lower cost. No obligation—just mail the coupon for details. *Stemco Corporation, Cleveland 16, Ohio.*

MAIL THIS COUPON

Stemco Corporation,
Cleveland 16, Ohio

Please send details and arrange for demonstration of RAMSET FASTENING SYSTEM and how it will save time and cut our costs. HP

Name_____

Address_____

City_____

visor, Sturtevant Div., Westinghouse Electric Corp. ..... **Harry E. Lewis** has been assigned to the foreign and export department, **Worthington Pump and Machinery Corp.**, Harrison, N. J. **George Bourque** has succeeded him as works comptroller at the Holyoke, Mass., works.

Penn Electric Switch Co., Goshen, Ind., has named **Howard C. Shilling** manager of the Chicago district office, succeeding **E. M. Smith.** Mr. Shilling joined the company in 1941 and covered the Chicago territory until he was appointed manager of the Moline, Ill., district office. He served as manager of the Goshen district before leaving the firm last year to enter the wholesaling field. ..... **Ben M. McDougall** has joined **Kennard Corp.**, St. Louis, as assistant sales manager. He was formerly laboratory test engineer and a sales representative in the southeast area for Alco Valve Co. and more recently was manager of Crown Insulation Co.

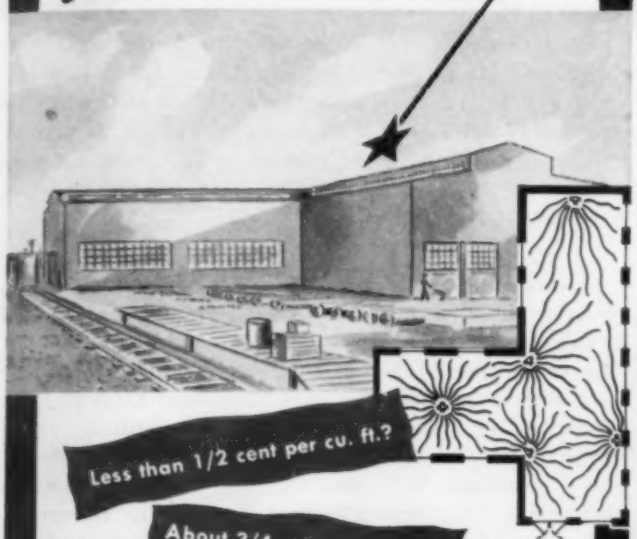
James M. Ashley, director of public relations, Libbey-Owens-Ford Glass Co., has been re-elected president of the **Producers' Council**, national organization of building product manufacturers. ..... **Richard T. McCauley** is responsible for the initiation of all activity of **Herman Nelson Corp.**, Moline, Ill., with respect to United States and foreign government purchases of its products. For the past four years he has served as a public relations officer in the office of the Secretary of the Air Force and Secretary of Defense.

The **Multi-Vent Div., Pyle-National Co.**, Chicago, has appointed the following new representatives: **J. P. Ashcraft Co.**, 2826 Fondren Dr., Dallas; **R. M. Brand Co.**, Tucson; **Joseph P. Maguire**, 615 Commercial Pl., New Orleans; **E. H. Norton**, 616 Lloyd Bldg., Seattle; **Pacific Scientific Co.**, 1430 Grande Vista Ave., Los Angeles; **J. Arch Colbrunn**, 1605 McLemore Ave., Memphis; **Francis J. Fallquist**, 507 Kuhn Bldg., Spokane; **Thomas E. Price**, 11 Oak Court, Houston; **Northwest Furnace Supply Co.**, 911 Northwest Hoyt St., Portland, Ore.; and **S. C. Pardee**, Sharon Bldg., San Francisco.

Wm. G. Boales & Associates, Detroit, is now representing **Cooper Alloy Foundry Co.**, Hillside, N. J. The Michigan firm has handled an extensive line of heating, cooling, and related industrial products for the past 20 years and has recently named **Wayne Karber**, a graduate in engineering, as sales engineer in charge of cast stainless steels. ..... The **Chicago Board of Education** has appointed **Thomas J. Brett**, one of seven school supervising engineers, as acting chief engineer to succeed the late **John Howatt.** He had been Mr. Howatt's chief assistant for the last two years.

Nominees for national office in the **American Society of Refrigerating Engineers** for 1950 are: **John G. Bergdoll, Jr.**, vice president and general works manager, York Corp.—for president; **Paul B. Christensen**, vice president, Merchants Refrigerating Co., and **Edward Simons**, consulting engineer—for vice presidents; **Richard C. Jordan**, professor of mechanical engineering, University of Minnesota—for treasurer; and for directors with three year terms—**Frank H. Faust**, manager, commercial engineering divisions, air conditioning dept., General Electric Co.;

How much would you say it cost to heat this Plant?



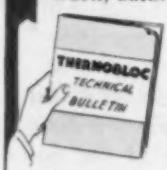
Less than 1/2 cent per cu. ft.?

About 3/4 cent per cu. ft.?

9/10ths of a cent per cu. ft.?

You wouldn't think there could be such a spread in the cost of heating a simple industrial structure—but an actual installation* figured out just that way! Coal fired pot bellied stoves cost 9/10ths of a cent per cu. ft.—additional boiler capacity, with necessary piping and diffusers figured at about ¾ cent per cu. ft. THERMOBLOCS to heat the structure cost .0044. All figures were based on a 15 year write off!

It's easy to see why THERMOBLOC presents such fundamental economy. Direct heat transfer, direct heat distribution—elimination of piping and duct work, automatic operation—all add up to economy.



PROVE IT YOURSELF!

To assist you in figuring heating requirements, THERMOBLOC has prepared a new bulletin containing engineering data tables and standard information on heating constants, building materials, etc. With this bulletin you can estimate the number and size of THERMOBLOCS to supply the necessary heat load, and you can make accurate forecasts of fuel consumption!

**Facts are from an article titled "Direct-Fired Unit Cuts Heating Costs"—printed in OPERATING ENGINEER, Nov., 1948. Copy of the article will be sent upon request.*

THERMOBLOC DIVISION
PRAT-DANIEL CORPORATION
Manufacturers of P.D. Fan Stacks and Dust Collectors

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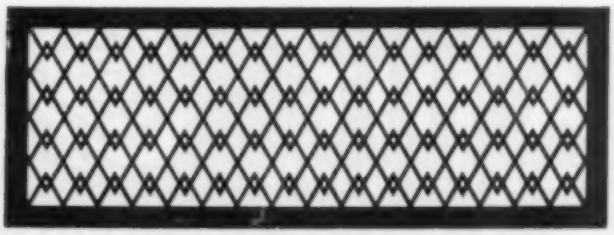
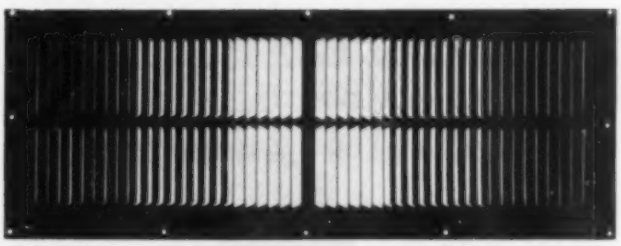
East Port Chester, Conn.



A Hendrick grille design (M No. 9) suitable for use with deflecting unit in Bulator combination.

The Perfect Combination
of a deflecting unit and ornamental grille



Vertical deflecting vanes, showing how the vanes may be set to produce any desired air stream pattern.

Now you can secure all the advantages of an adjustable deflecting vane grille—*to control air flow*—and an ornamental grille—*to harmonize with the decorative scheme*—by specifying the new dual-unit Hendrick BULATOR.

For this combination of deflecting unit and ornamental grille there's a wide selection of Hendrick ornamental grille designs which have ample open area so the air throw and spread are not appreciably affected.

Mounted just behind the ornamental grille, the deflecting unit is not noticeable, and its vanes are adjustable so that the air flow can be deflected to right or left, up or down, or in a combination of directions.

Write for full information.

Perforated Metals
Perforated Metal Screens
Architectural Grilles
Mitco Open Steel Flooring,
"Shur-Site" Treads and
Armorgrids

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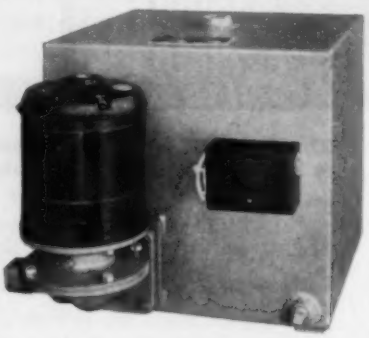
Manufacturing Company
48 DUNDAFF STREET, CARBONDALE, PA.
Sales Offices in Principal Cities

A HIT from the start! the Sterlco 4100 SERIES CONDENSATION PUMP

The popularly priced 4100 Series pump, the newest addition to our complete line of condensation pumps, has really "hit the bullseye" with users. Its compactness and neat appearance, its simplicity and sturdiness of construction, its conservative rating for handling emergency overloads, have all contributed to its enthusiastic acceptance.

Before you specify or buy a small condensation pump, be sure to find out about the many quality features of the 4100! The coupon below will bring full information.

The 4100 Series is particularly attractive as a stock unit. Wide range of capacity and pressure ratings in one unit eliminates stocking number of pumps. Requires less than 2¾ sq. ft. of storage space, stacks conveniently.



Sterlco 4100 Series Condensation Pump with steel tank. ¼ H.P. capacity up to 6000 sq. ft. E.D.R. at 20 lbs. Also ½ H.P. unit for 8000 to 14000 sq. ft. at 20 lbs.

New Sterlco 4200 Series unit, for jobs where cast iron tank is preferred. Same ¼ H.P. pump as the 4100—for 6000 sq. ft. maximum capacity. 20 lb. pressure. Also ½ H.P. unit for 8000 to 14,000 sq. ft. at 20 lbs.

All STERLCO products are distributed through leading heating and plumbing wholesalers.

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GENTLEMEN: Please send me complete information on Sterlco Condensation Pumps.

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Firm.....

Address.....

City.....Zone.....State.....

LOOK AT SOME OF THESE OTHER

Sterlco PRODUCTS

Arthur J. Hess, partner, Hess, Greiner & Polland; and **George K. Iwashita,** manager of product research material and process, Seeger Refrigeration Co.; **Rowland H. Lock,** general manager, J. H. Lock and Sons, Ltd.; and **Hermann F. Spoehrer,** vice president, Sporlan Valve Co.

R. B. Yirak is now secretary of **Steel Products Engineering Co.,** Springfield, Ohio. He has been associated with the coal and stoker business for more than 25 years and had been assistant secretary. ..... **R. C. Anderson,** formerly southern district manager for A. O. Smith Corp., is the new Houston, Tex., distributor for **Bryant Heater Div., Affiliated Gas Equipment, Inc.,** Cleveland. He is located at 3215 Locke Lane, Houston. ..... **The National Association of Credit Men** has elected **John G. dePass** vice chairman of its plumbing, heating, refrigeration, and air conditioning industry committee. He is general credit manager and assistant secretary of National Radiator Co.

Edward Latta has returned to **Redmond Co., Inc.,** Owosso, Mich., as a special project engineer. From 1939 to 1946 he served with the company and during the last year of that period he was chief engineer. More recently he had been chief engineer of Universal Electric Corp. Other men assigned to the new project engineering section of Redmond are **George C. Morris, John T. Howes,** and **F. Richard Merriam.** ..... **Thompson Equipment Co.,** New Orleans, headed by **Horace A. Thompson, Jr.,** is now representing **Hammel-Dahl Co.,** Providence, R. I., in central and eastern Louisiana, southern Mississippi, and northwest Florida.

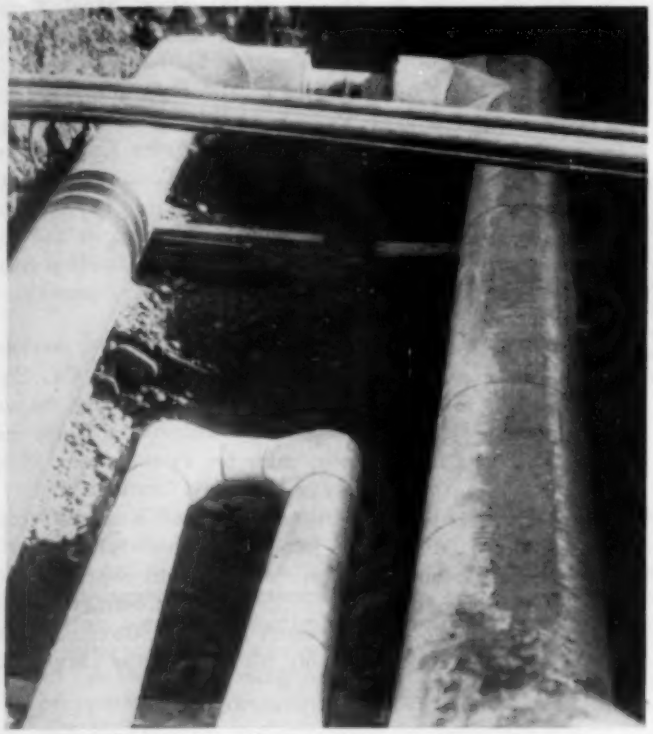
Clapp Instrument Co., Webster, Mass., has elected **Arthur S. Brown** president. He was formerly associated with Manning, Maxwell & Moore, Inc., Mason-Neilan Regulator Co., and Fischer & Porter Co. He has been acting as general manager of Clapp for several months. **Charles Clift** has been appointed treasurer, and **T. W. Lanman** is the new vice president. ..... **A. E. Hess** has resigned as managing director of **Oil-Heat Institute of America, Inc.,** and has joined the marketing and distribution division of **Lustron Corp.** Well known in the heating industry, Mr. Hess has supervised OHI activities during the past five years.

Iron Fireman Mfg. Co., Cleveland, has named **Reidar A. Norbom** industrial sales engineer in the eastern division. For four years he was district manager of the Boston office of Orr & Sembower, Inc. ..... The **Dayton Sales Executive Club** has started its 1949-50 series of meetings under the direction of **Robert A. Mayne,** president, Mayne Products Co. Making better salesmanship is the subject for the year's programs.

Dr. Nicol H. Smith, director of the division of chemical engineering and physics, **Franklin Institute Laboratories for Research and Development,** Philadelphia, has recently been appointed director of research operations. .... The **American Society of Refrigerating Engineers** has formed two new technical subcommittees. Subcommittee A-11 on heat pumps, under the chairmanship of **Charles H. Coogan, Jr.,** head of the mechanical engineering dept., University of Connecticut, will coordinate

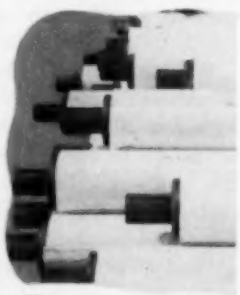
D.I.P.
REG. U. S. PAT. OFF.

Pre-Sealed Protection Built-In at Factory



- and Low Cost Installation Saves Time, Trouble and Work

Wherever you have a problem of conveyance of hot-water, steam or refrigerants, you can specify DURANT INSULATED PIPE with complete confidence in securing long range economy.



The combination of the best quality moulded insulation completely protected by non-porous asphalt merits the preference of architects, engineers, and contractors who have investigated the scientifically engineered construction methods employed in the fabrication of D. I. P.

Economy starts with installation. D. I. P. requires no tile, masonry, or sub-drains. Field joint work is simple, yet corrosion and electrolysis are eliminated. Low maintenance costs result in further economy over long years of dependable service.

Complete detailed information and engineering data on standard piping and special fittings will be furnished on request. Our representatives in principal cities are at your service.

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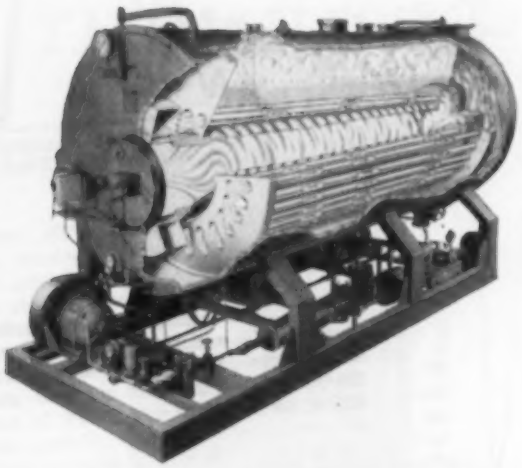
1015 Runnymede St. East Palo Alto, California

THIS airliner's streamlined for speed



This AMESTEAM GENERATOR is Streamlined for Saving

This cut-away view of an AMESTEAM Generator shows the three full-length-of-boiler passes arranged to provide a smooth, even flow of gases through the Unit — insuring more efficient conversion of your fuel BTU's into steam, with resultant low stack temperatures, high CO_2 and a guaranteed thermal efficiency of 80%.

And it's fast, too — full steam pressure in 20 to 30 minutes.

Buy an AMESTEAM Generator — substitute its consistently smooth daily operation for your present inefficient boiler room conditions.

AMESTEAM Generators are available from 10 h.p. to 400 h.p.; for pressures from 15 lbs. to 200 lbs.; and for firing with Light Oil, Heavy Oil, Straight Gas, Light Oil-Gas or Heavy Oil-Gas Combinations. For full information —

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CASE REPORT No. 53 from a series of actual cases recording the successful application of "BEAR" DY-NAMIC BALANCING to Industry

High Speed Centrifugals

DY-NAMICALLY BALANCED

ACCURACY demanded for BIRD CENTRIFUGAL FILTERS

BIRD MACHINE CO., SOUTH WALPOLE, MASS., reports: "Bird Centrifugal Filters are used in practically every industry where it is necessary to separate solids from liquids. These machines can handle solids as fine as a fraction of a micron, or tonnages as high as 60 tons per hour ... they must be precision-built. All rotating parts are Dy-Namically Balanced on our 'Bear' Machine Model 400-R. In some parts up to 25 lbs. of unbalance are indicated, but the 'Bear' Machine attains both Dy-Namic and Static Balance in one quick, simple operation. Using the 'Bear' Method, we obtain accurate results as well as efficient production."

Here, the operator is adjusting the micrometer dial of the "Bear" Machine at Bird Machine Co. in order to determine the amount of Dy-Namic and Static Unbalance.

"BEAR" METHOD

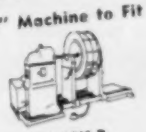
• Improves Performance
• Speeds Up Production
• Sells MORE of YOUR Products

The "Bear" Method of Dy-Namic Balancing is making its weight felt in every phase of industry by helping to increase sales appeal and adding value to hundreds of products ... from tiny electric motors to washing machines and railroad wheels. By the revolutionary "Bear" Method, both Dy-Namic and Static Balance are attained in one time-saving, efficient operation. The Dy-Namic Balancing operation need not interrupt or slow down routine production and may be installed with low initial capital investment.

Investigate the many production advantages and sales gains made possible by "Bear" Dy-Namic Balancing—GET ALL FACTS in this FREE 22-pg. DY-NAMIC BALANCING MANUAL. Write "Bear" Mfg. Co., Dept. H-16, Rock Island, Ill.

There's a "Bear" Machine to Fit Your Needs!

Model 337-V
Max. Capacity 6 lbs.



Model 340-R
Max. Capacity 100 lbs.

Model 375-R
Max. Capacity 2,000 lbs.

(See Dy-Namic Balancing Manual for other models)

3031

"BEAR" STATIC AND DY-NAMIC BALANCING MACHINES

Trade Mark Registered U. S. Patent Office

its efforts with other industry groups. Subcommittee B-3, Miscellaneous, headed by **D. E. Rutishauser,** Hussmann Refrigeration Co., will be assigned functions not under the jurisdiction of existing technical committees.

S. C. Wood and **K. P. House** are new assistant managers of the tubular sales division, **Wheeling Steel Corp.,** Wheeling, W. Va. Mr. Wood has had many years of experience in the division, and Mr. House was formerly with the Chicago sales office of the firm and has been Pacific coast manager. ..... As a new staff executive of **Winchester Repeating Arms Co., Div. of Olin Industries, Inc.,** New Haven, Conn., **James C. Hartley** is handling special assignments for the regional manager. He is a nationally known metallurgist and had been vice president and general manager, Barium Steel and Forge, Inc.

E. A. Tracey is now vice president and general manager of the air conditioning division, **Mitchell Mfg. Co.,** Chicago. Prior to his association with the company, he was president of Majestic Radio and Television Corp. and vice president in charge of sales for Zenith Radio Corp. ..... **Dudley K. Bailey,** until recently a member of the sales engineering staff of Weston Electrical Instrument Corp., has joined the sales organization of **Beedle Equipment Co.,** Cincinnati, district representative for Weston products. ..... **Commercial Filters Corp.,** Boston, has opened a new midwestern sales office at 603 W. Washington Blvd., Chicago. **Walter H. Magee** is in charge.

John L. Collyer, president, B. F. Goodrich Co., has been elected a trustee of the **Alfred P. Sloan Foundation** which was founded in 1936 and since that time has expended $9 million in grants. ..... **Perfex Corp.,** Milwaukee, has appointed **Charles H. Burkhardt** director of field education. He will work with dealers and distributors preparing information and material on service, application, and installation.

Williams Oil-O-Matic Div., Eureka Williams Corp., Bloomington, Ill., has made a number of personnel changes. **J. H. Shoemaker,** formerly district sales representative in New England, has been named assistant to the general sales manager. **Ralph H. Ellerbrock,** associated with the corporation and its predecessor company for 25 years, is now manager of brand name sales. He succeeds **W. A. Matheson, Jr.,** who recently resigned. **William A. Sayler** and **Robert Davis** are now district sales representatives in Ohio and upper New York states respectively. Mr. Sayler was formerly divisional sales manager for the heating division, Wayne Home Equipment Co., and Mr. Davis had been a sales engineer for Clark Industries.

Distributor sales of **Flexible Tubing Corp.,** Branford, Conn., are being managed by **A. H. Loux** who has been a direct factory representative for the company in northern New York state and surrounding territory. ..... **Tex-Rock Insulation Mfg. Co.,** Temple, Tex., has opened a new branch office at 162 Leslie St., Dallas, with **W. J. Briggle, Jr.,** in charge. In addition to handling its own insulation, both the Dallas and Temple branches act as distributors for related products in the cooling, heating, and insulation field.

IMPROVE YOUR KNOWLEDGE OF AIR CONDITIONING

SEND TODAY FOR

Samuel R. Lewis'

"AIR CONDITIONING FOR COMFORT"

288 Pages—6¼" x 9¼"—Illustrated—

Cloth Bound—$2.00

$2.00

Here is a book that presents—in simple, readily understandable form—every kind of information necessary for an accurate and thorough knowledge of air conditioning principles, equipment and practices. Written by S. R. Lewis, a widely-known consulting engineer who has been active in air conditioning work for more than thirty years, it deals with all angles of the air conditioning subject from the practicing engineer's viewpoint. The designing procedures explained in the book are, for example, in every detail the same procedures employed today by the author's own organization.

Featuring this third edition are chapters on phases of the subject not previously treated, including noise control, air conditioning measurements, air conditioning standards, fire protection codes and operating suggestions. Several complete examples of correct design procedure are given, together with forms for recording the design data, the proper filling-in of which are explained step-by-step.

The chapter devoted to Psychrometry presents nineteen different formulas for psychrometric calculations. In illustrating the correct use of these formulas, Mr. Lewis applies both the psychrometric tables and charts in order to render both devices thoroughly understandable.

OF VALUE BOTH AS A REFERENCE AND TEXT

Engineers in air conditioning will find the "Air Conditioning for Comfort" invaluable as a reference book, while salesmen, students, and others may rely on it to give them a clear knowledge of fundamentals, and of air conditioning methods and equipment.

Send for a copy today. We know you will consider this volume one of the most readable and complete books on the air conditioning science you have yet seen. You will risk nothing in ordering a copy, for you will be privileged to return it for a refund if for any reason it should prove unsatisfactory. Use the coupon at the right to order your copy now.

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CHAP. 2—Heat and Air
CHAP. 3—Air Conditioning Standards
CHAP. 4—Air Conditioning and the Human Body
CHAP. 5—The Psychrometric Table and Chart
CHAP. 6—Humidity Controllers and Control
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CHAP. 18—Noise and Its Control
CHAP. 19—Air Conditioning Instruments and Measurements
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10" x 16" Psychrometric Chart..... Inside Back Cover

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Enclosed is $2.00 for a copy of AIR-CONDITIONING FOR COMFORT. If this book should prove unsatisfactory, I will return it within 15 days for a refund.

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STREET.....

CITY & STATE.....

NEW BOOKS & REPORTS

Surveys Available Knowledge of Odor Perception and Control

Early civilizations within the limits of available knowledge developed appropriate answers to the odor-nuisance problem, resorting to such expedients as incense, perfume, and wood burning to combat the unpleasant odors prevailing in their daily living. It is only within the last century, however, that odor control has been investigated by scientific methods. Odors, Physiology and Control surveys the available knowledge of odor perception and odor control—problems so intangible that their analysis still proceeds from subjective human perception. The authors are Carey P. McCord, Industrial Health Conservancy Laboratories, and W. N. Witheridge, ventilation consultant, General Motors Corp.

Dividing the discussion into three major parts, the anatomy and physiology of olfaction, the odor problem, and the control of odors, the authors have succeeded in producing an extremely useful summary, placing considerable emphasis on the problem of odor nuisances. Moreover they have clearly indicated the areas that require further controlled investigations.

In a discussion of odor control, the authors distinguish between odor masking and cancellation-counteraction, although the evidence for such a distinction may be open to question. The chemical, physical, and electrochemical theories of odor perception, with the exception of the infra-red theory of Beck and Miles and the electro chemical theory of Henry Eyring, are briefly presented. It is evident that no single pattern of chemical structure contributes to the characterization of odors.

The industrial phases are given special treatment under such headings as meteorology of odors, air and stream pollution, disposal of solid, trade waste and sewage, and malodorous gases and vapors. Chemical and physical methods of odor destruction and odor masking are presented in some detail.

Activated carbon is discussed in great detail, even to the extent of reproducing Munkelt's tables on the retentivity of carbon for gases and vapors in standard air conditioning installations, as well as detail drawings of modern appliances used with carbon.

The book ends with a plea for objectivity in the conducting of odor surveys and a discussion of the legal rights of citizens confronted with odor nuisances.

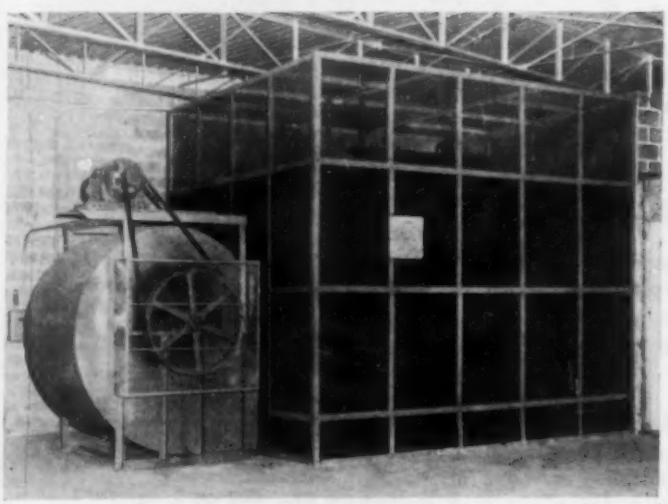
The text is both authoritative and detailed. It should be of inestimable value to public health officers, sanitary and air-conditioning engineers, industrialists, and to the research worker, as well as to the interested layman.

It is unfortunate that the footnoted journal references fail to distinguish between the number of the volume and the number of the page. Footnotes are for the most part accurate, although it was noted that the quotation on page 193 is not to be found from the reference cited.

Terminating the book is an extensive bibliography, estimated at 3500 items, which selectively reviews the literature on odors and olfaction from the early 19th century into the middle of the 1940's. The authors appreciate such shortcomings of the bibliography as the inaccuracies resulting from the inability to consult original works, the

JOHNSON DIRECT-FIRED AIR-ROTATION HEATERS

ARE DOING A BETTER HEATING JOB AT LESS COST



200,000 to 5,000,000 BTU in single units.

The Johnson System of Air-Rotation (Pat. Apl'd. For) circulates large volumes of warm air to all parts of an open building without piping or ducts, and heats the building faster than any other method. It keeps the warm air at the "working level", which means comfortable working conditions throughout the plant, regardless of outside weather, and it means warm, dry floors.

Fast heat recovery is assured even when large loading doors are frequently opened and closed.

Johnson Direct-Fired Heaters are fully automatic in operation, using either light or heavy oil, coal with stoker firing, or gas.

The Johnson Heater is revolutionary in design. Multiple banks of criss-cross air heating tubes are placed directly over the fire for maximum heat transfer from the fire to the heated air delivered into the building.

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Vapor seal	simple	complicated or difficult to install
Structural strength	very good	some non-existent, others fair
Weight	very light	varies
Resilience	considerable	from fair to complete rigidity
Settling	none	a serious problem
Supporting structure	none	elaborate
Fire resistance	good	excellent to poor
Vermin resistance	good	excellent to very bad
Odor absorption	none	some are regular sponges

incomplete listing of authors named in the reference, and the lack of organization of the bibliography in relation to the text. Despite the criticisms which might be raised against the bibliography, this compilation and the book as a whole represent a tremendous effort on the part of the authors to produce a very much needed volume. This text brings into focus a previously disorganized mass of superstition, conjecture, and scientific data and thus provides a starting point for a scientific attack on a grossly neglected field.

Published by McGraw Hill Book Co., 330 W. 42nd St., New York 18, this 6 x 9 in., clothbound text contains 405 + x pages and is priced at $6.50.—Reviewed by William V. Consolazio, head, biochemistry branch, Office of Naval Research, Department of the Navy.

ASME Revises Boiler Construction, Unfired Pressure Vessel Codes

During the past 10 months, the boiler code committee of the American Society of Mechanical Engineers has recommended a number of changes and additions to the 1946 Boiler Construction Code. These include revisions of the preamble of the low pressure heating boiler code in order to more clearly define the capacity limits of hot water supply boilers, revisions of the classification of welding operators, amplification of stress tables P-7 and U-2 to assign allowable working stresses to SA-213 materials in grades T7 and T9, revised formulas for bolted flange connections, provisions for construction of unfired steam boilers in both the power boiler and unfired pressure vessel codes, adoption of working stresses for use with copper, nickel, and aluminum materials and their alloys, and revisions of 36 existing material specifications and inclusion of six new specifications.

The new 1950 Unfired Pressure Vessel Code provides rules for the design, fabrication, and inspection (during construction) of carbon and low alloy steel vessels, including those operating at temperatures below −20 F, vessels fabricated primarily by fusion welding, and vessels or parts of vessels fabricated by riveting or by closely fitted bolting.

Some of the important changes of the 1950 code, as compared with the earlier edition, are the clarification of the rules continued from the former code, increased design stresses, completely revised rules for reinforcement of openings, broader inspection rules, requirements for capacity certification of pressure relief devices, a sliding scale of joint efficiencies for welded pressure vessels, and provisions for the design of vessels under external pressure at elevated temperature.

The 1949 edition of the boiler code (including addenda published in 1947, 1948, and 1949) and the 1950 edition of the unfired pressure vessel code are scheduled for publication this month. Both may be obtained from ASME headquarters, 29 W. 39th St., New York 18. The complete boiler code (nine sections and binder) is priced at $14.50. Individual sections may be purchased separately as follows: power boiler code including rules for inspection, $2.75; unfired pressure vessel code, $2.25; low pressure heating boiler code, $1.00; miniature boiler code, 75 cents; welding qualifications, 90 cents; material specifications, $5.00; and suggested rules for power boilers, $1.25. The 1950 pressure vessel code is priced at $3.50.

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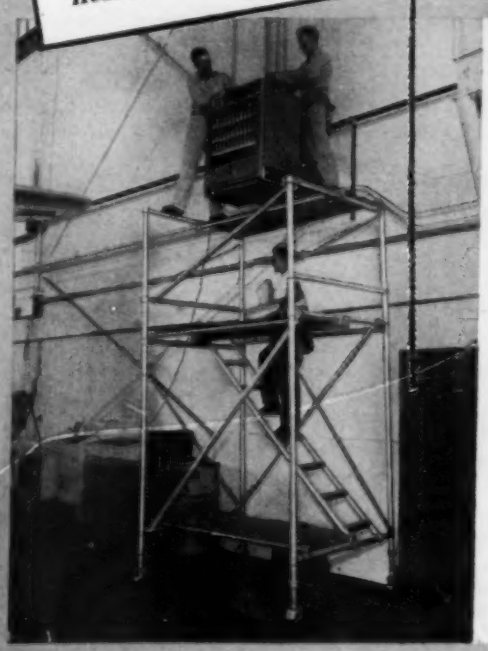

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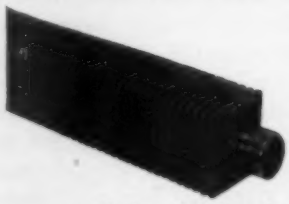

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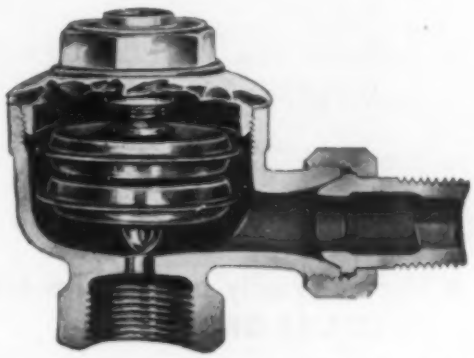

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ASRE Proposes Standard for Rating, Testing Refrigerant Compressors

Proposed Standard No. 23-R of the American Society of Refrigerating Engineers prescribes methods of rating and testing refrigerant compressors and provides for both absolute and comparative evaluation of performance and capacity under specific, standardized conditions.

The standard classifies compressors as to type, lists and defines the terms used, establishes conditions and methods of test, and lists the conditions which should be specified for application ratings under operating conditions other than those designated for ASRE standard ratings.

Types of compressors covered are single stage reciprocating, multi-stage reciprocating, centrifugal, and rotary type units. In recognition of variations that exist in practice for various applications, 13 groups of proposed standard rating conditions are recommended. Standard test methods consist of a primary test and a simultaneous confirming test, the results of which, in general, must be within 3 percent of the primary test results. Primary test methods set forth in the standard include brine tank calorimeter, concentric tube calorimeter, secondary refrigerant calorimeter, flooded system primary refrigerant calorimeter, dry system primary refrigerant calorimeter, and the gaseous refrigerant flow meter method. Details on confirming test methods are given for liquid refrigerant quantity, liquid refrigerant quantity meter and flow meter, condenser, and compressor heat balance methods.

Copies of the proposed standard may be obtained from ASRE headquarters, 40 W. 40th St., New York 18, for 50 cents.

Other Books and Reports Received

A Study of a Baseboard Convector Heating System in a Test Bungalow, by P. R. Achenbach and E. M. Tierney —Building Materials and Structures Report BMS115, published by the U. S. Department of Commerce, National Bureau of Standards and available from the Superintendent of Documents, U. S. Government Printing Office, Washington 25, D.C. for 15 cents. Tests were made in a four room basementless structure to determine temperatures produced at various levels above the floor with a baseboard heating system using finned piping adjacent to all exterior walls and at the baseboard level. Great comfort was noted near the floor.

Mineral Wool Insulation for Heated Industrial Equipment, Commercial Standard CS117-49—Issued by the Commodity Standards Div., National Bureau of Standards. Printed copies are available from the Superintendent of Documents, U. S. Government Printing Office, Washington 25, D.C. for 15 cents. The standard provides requirements and test standards for mineral wool in the form of loose or granulated wool, felts, industrial batts, blocks or boards, pipe insulation, and insulating cement for insulating heated industrial equipment. It also covers thicknesses required for various operating temperatures and recommended methods of installation.

Heat Transfer and Fluid Flow—Twenty-two technical papers, presented at the recent meeting of the Heat Transfer and Fluid Mechanics Institute, have been published in the form of a 280 page book by the American Society of Mechanical Engineers, 29 W. 39th St., New York 18. The price is $5.00.

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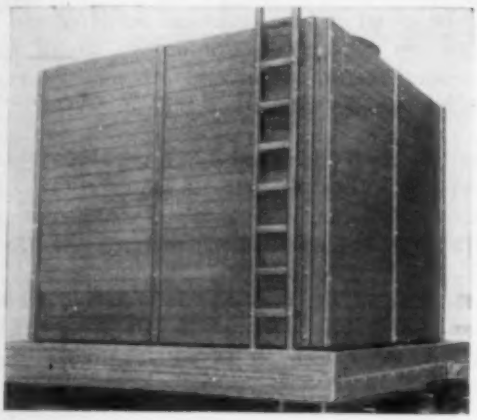

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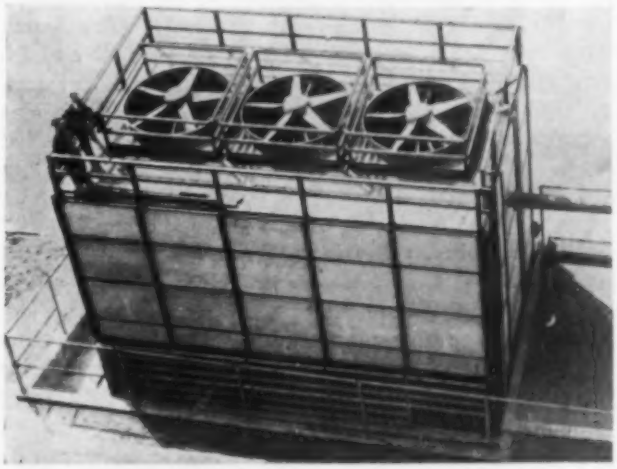

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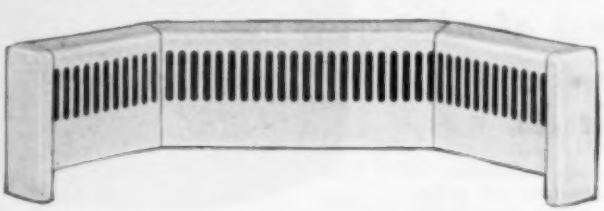

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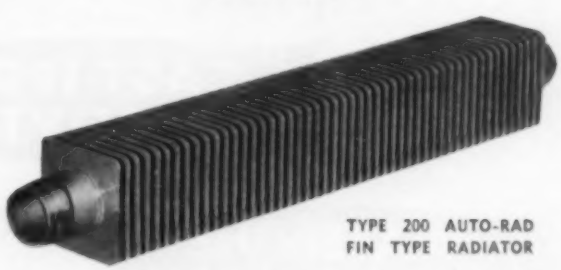

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MEETINGS & CONVENTIONS

Refrigeration & Air Conditioning Contractors Association (formerly National Association of Refrigeration Contractors), 228 N. La Salle St., Chicago 1—The fourth annual convention, Atlantic City, November 13 and 14.

Papers to be given include Air Conditioning Imagineering, by C. S. Stackpole, vice president and general sales manager, Airtemp Div., Chrysler Corp. Others will deal with sales, merchandising, and antitrust laws concerning access to markets.

Sixth All-Industry Refrigeration and Air Conditioning Exposition, 1346 Connecticut Ave., N.W., Washington 6, D. C.—November 14 to 18, Atlantic City Auditorium, sponsored by the Refrigeration Equipment Manufacturers Association. Headquarters for cooperating associations are: Traymore hotel, REMA and exhibitors; Claridge hotel, Refrigeration Equipment Wholesalers Association; Ambassador hotel, Refrigeration Service Engineers Society and Refrigeration and Air Conditioning Contractors Association; Chalfonte-Haddon Hall, Air Conditioning and Refrigerating Machinery Association and National Electrical Manufacturers Association.

The exposition will be open to contractors, dealers, and service engineers only on November 14.

American Society of Mechanical Engineers, 29 W. 39th St., New York 18—Annual meeting, November 28 to December 2 at the Hotel Statler, New York City.

Over 270 speakers will deliver some 200 papers, and there will be 78 technical sessions sponsored by 22 professional divisions and committees of the society. Some of the many fields to be covered include fuels, heat transfer, hydraulics, industrial instruments and regulators, and process industries. At a luncheon of the heat transfer division on November 29, F. C. Hottel, Massachusetts Institute of Technology, will speak on uses of solar radiation.

The Society's spring meeting is scheduled for April 12 to 14, Hotel Statler, Washington, D. C.

American Society of Refrigerating Engineers, 40 W. 40th St., New York 18—1949 annual meeting, December 5 to 7, Edgewater Beach hotel, Chicago. Twelve technical papers, to be given at four technical sessions, and three conferences are scheduled. Conference topics include water re-use in refrigeration and air conditioning and room conditioners in multi-room buildings.

National Society of Professional Engineers—The 1949 annual meeting, December 8 to 10, Houston, Tex. Additional information may be obtained from the NSPE office, Houston Engineers Club, 2615 Fannin St., Houston.

National Warm Air Heating and Air Conditioning Association, 145 Public Square, Cleveland 14—The 36th annual convention, December 14 and 15, Hotel Cleveland, Cleveland.

American Meteorological Society, 5 Joy St., Boston 8—The 30th annual meeting, January 3 to 6, St. Louis. An

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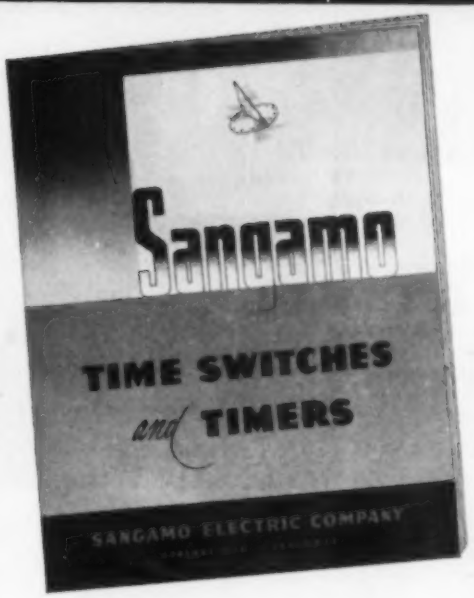

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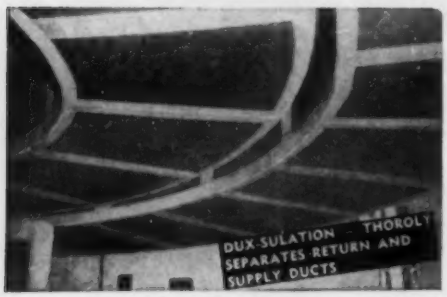

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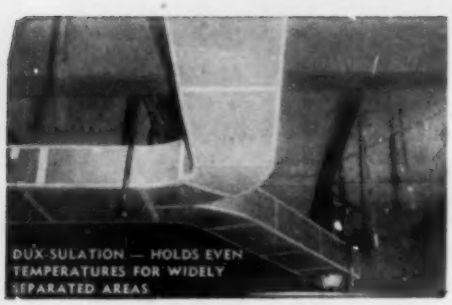

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Enforcement Principles—Desirable Requirements and Standards, by J. H. Carter, Smoke Commissioner, City of St. Louis

General Effects of Air Pollution on Health—Allowable Conditions, etc., by G. D. Clayton, U. S. Public Health Service

Industrial Accomplishments—Available Equipment; General Approach to the Problem, by C. A. Gosline, E. I. du Pont de Nemours & Co., Inc.

Dispersion of Stack Gases in Stable Atmospheres, by M. L. Barad, New York University

Meteorological Instruments Useful in Air Pollution, by Dr. Michael Ference, Jr., Evans Signal Laboratory

Instruments for Air Pollution Measurement, by W. C. L. Hemeon, Industrial Hygiene Foundation of America, Inc.

Plant Maintenance Exposition and Conference, January 16 to 19, Cleveland Auditorium—Exhibits and conference subjects will deal with air conditioning, heating, ventilating, building supplies, electrical equipment, employee relations, instruments, lubricants, paints, materials handling, power generation and transmission, welding, etc.

Advance registration cards may be obtained from Clapp & Poliak, Inc., 341 Madison Ave., New York 17, exposition manager. For hotel reservations, write to the Plant Maintenance Show Housing Bureau at the Cleveland Convention Bureau, Terminal Tower, Cleveland.

American Society of Heating and Ventilating Engineers, 51 Madison Ave., New York 10—56th annual meeting, Adolphus and Baker hotels, Dallas, Tex., January 23 to 26. For additional information see the ASHVE Journal Section in this issue.

Southwest Air Conditioning Exposition of the International Heating & Ventilating Exposition—State Fair Grounds, Dallas, Tex., January 23 to 27. As in past years the show will be managed by International Exposition Co., Grand Central Palace, New York City, and will be held under the auspices of the American Society of Heating and Ventilating Engineers which is represented by an advisory committee headed by A. E. Stacey, Jr., president of ASHVE. The show is expected to fill the Hall of Agriculture and the Poultry Building. For details see the ASHVE Journal Section in this issue.

American Society for Testing Materials, 1916 Race St., Philadephia 3—Committee week and spring meeting, Hotel William Penn, Pittsburgh, February 27 to March 3.

The 53rd annual meeting and 9th Exhibit of Testing Apparatus and Related Equipment will be held June 26 to 30, Chalfonte-Haddon Hall, Atlantic City.

National Association of Corrosion Engineers, 919 Milam Bldg., Houston 2—The 6th annual conference, St. Louis, April 4 to 7. Symposia under consideration include round table discussions on general corrosion problems and corrosion in connection with pipe lines, refrigeration and air conditioning, and the food industry.

National Oil Heat Exposition, sponsored by the Oil-Heat Institute of America, Inc., 6 E. 39th St., New York 16.—The affair will be held in conjunction with the 27th annual meeting of OHI, the week of April 24, Commercial Museum, Philadelphia.

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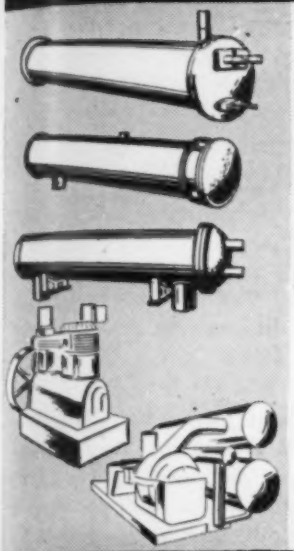

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MICHIGAN CITY, INDIANA, U.S.A.

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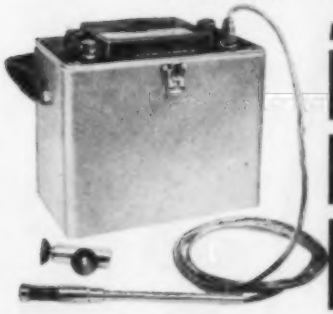

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AIR VELOCITY
FROM 10FPM to 6,000FPM

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FROM 30F to 155F

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FROM 0 TO 4 NEG. AND
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For Vertical Shell and Tube Ammonia Condensers

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RECENT TRADE LITERATURE

For your convenience in obtaining copies of these bulletins, see coupon on page 185. If you write direct to the manufacturer, describe carefully what literature you want, as the number given first in each item is for use only when sending requests to Heating, Piping & Air Conditioning.

No. 7464: Air Cylinders—"Power Dome" nonrotating, double acting air cylinders are described and illustrated in a 12 page bulletin (PD-120) issued by Bellows Co., 222 W. Market St., Akron, Ohio. Various sizes and types of mountings are covered and dimensional drawings are included.

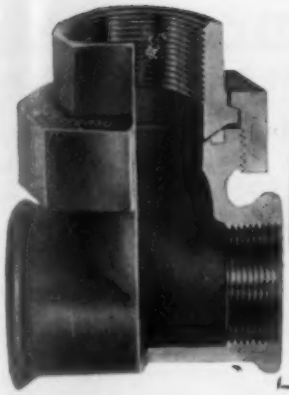
No. 7465: Air Filters for Paint Spray Installations—Research Products Corp., 1015 E. Washington Ave., Madison 10, Wis., has released a new pamphlet (Form 1105) telling how its permanent washable air filters are used for intake air of paint spray booths and how replaceable type arrestor pads are used for removing paint from exhaust air.

No. 7466: Air Vents, Other Heating Specialties—A new 12 page catalog of Dole Valve Co., 1933 Carroll Ave., Chicago 12, covers steam, hot water, and warm air heating specialties. Described and illustrated are vent valves for one pipe steam systems, quick venting return line air valves, vacuum valves, automatic vent valves for hot water heating systems, water mixing valves, flow control valves, and a thermostatic air control for use in forced warm air heating.

No. 7467: Arc Welding Award and Scholarship Program—Rules and conditions for the 1949-50 engineering undergraduate award and scholarship program are printed in a 24 page booklet available from James F. Lincoln Arc Welding Foundation, Cleveland 1. In addition to rules and conditions for participation, the booklet contains information on previous award programs, including descriptions of some of the winning papers. Suggested subject matter for student papers and a welding bibliography are also presented.

No. 7468: Ball and Roller Bearings—A new 112 page detailed catalog and data book covers the complete line of ball and roller bearings offered by Link-Belt Co., 307 N. Michigan Ave., Chicago 1. Identified as Book No. 2550, it covers design and construction features and gives list prices, weights, load ratings, and dimensions. A good deal of space is devoted to the selection of the proper bearing for the specific service. Selection examples and drawings of typical applications are included.

No. 7469: Baseboard Radiation—Constructional details, ratings, and conventional applications of "National Art" baseboard radiation are covered in Catalog No. 559 now available from National Radiator Co., Johnstown, Pa. The four page booklet illustrates and describes the installation of this component of a forced hot water heating system.



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AND HERE ARE JUST A FEW OF THE REASONS WHY!

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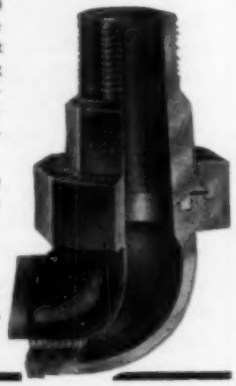

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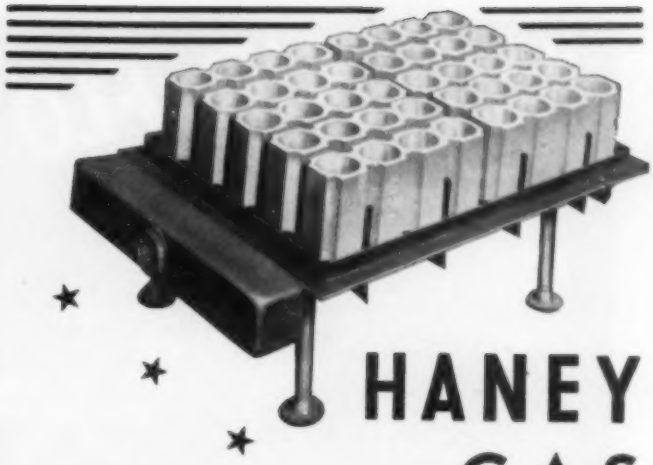

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Ask For Bulletin 24-1

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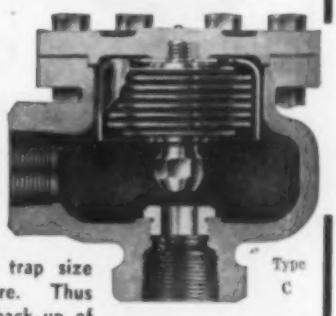
Patented Construction

Built For Hard Service

Nicholson Steam Traps

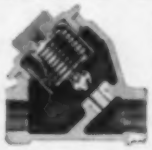
STOP BACK-UP of CONDENSATE

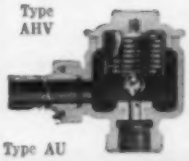
Operate on Lowest Temperature Differential



Type C

Repeated comparative tests by large users of traps show that Nicholson traps operate on lowest temperature differential: 5° to 15°, depending on trap size and pressure. Thus preventing back-up of condensate and keep equipment full of live steam. Installations have increased production of cooking kettles, for example, as much as 30%. Widely specified for preventing damage to thin gauges. Eliminate "cold blow" in unit heaters. Size ¼" to 2"; press. to 225 lbs.





Type AHV

Type AU

BULLETIN 1047.

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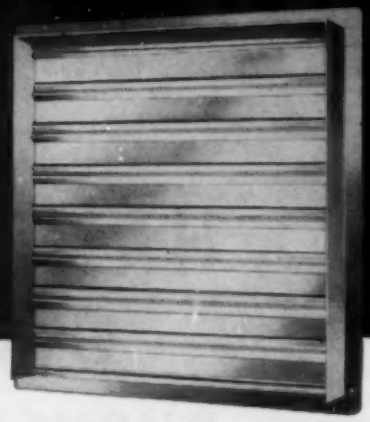

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FRONT VIEW—CLOSED

IT TAKES THE LOAD OFF THE FAN!

Aluminum louvers open fully, permitting capacity fan operation. New heavy reinforcement strip adds strength and long life to the louvers, assures quiet operation and perfect counterbalance, prevents rattling. Deep shroud protects shutter from high winds. Tie-rod, brackets and bearings inside frame, not exposed to weather. Special finish resists corrosion. Many other features.

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HEAT TRANSFER EQUIPMENT REMPE CO. Founded 1890

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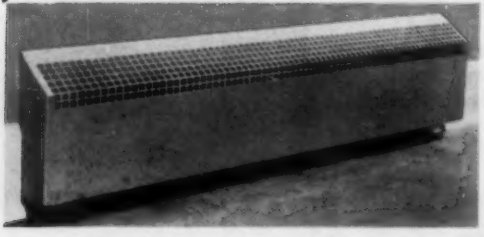

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Pipe and Fin Coils for Refrigeration and Air Conditioning.

No. 7470: Bending Machines—Information on bending machines for pipe, tube, reinforcement bars, sections, and structural shapes is contained in Bulletin 49 of Wallace Supplies Mfg. Co., 1304 Diversey Parkway, Chicago 14. Both hand operated and hydraulic machines are covered, as are special machines for specific uses. The pocket sized booklet also contains information on shears and cutters and fabricating tools for tubing.

No. 7471: Central Air Conditioning Equipment—Bulletin 3158B of Buffalo Forge Co., Buffalo 5, N. Y., gives detailed information on the firm's equipment for central air conditioning. Data are given on coil selection and installation and maintenance instructions are included.

No. 7472: CO_2 Analyzer—The analyzer of Charles Engelhard, Inc., 850 Passaic Ave., East Newark, N. J., is described in Bulletin 710. Offered for use with coal, gas or oil, the device not only measures the CO_2 content of flue gases by the thermal conductivity principle, but also indicates the temperature of the stack gases.

No. 7473: Combustion Control by Haze Density Measurement—Control equipment for industrial boilers and furnaces burning hydrocarbon fuels is described in Bulletin 103 of General Power Plant Corp., 381 Fourth Ave., New York 16. The nature of haze as a combustion index is discussed and devices described range from simple auxiliary controls for addition to present systems to complete "Robot-Eye" installations.

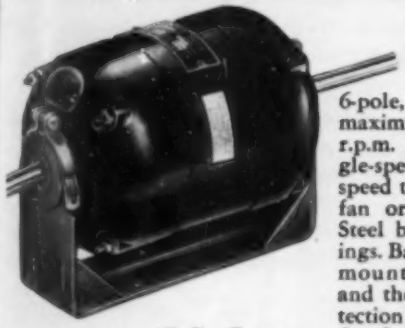
No. 7474: Concrete Drill—In connection with its "Konkrete Kore" drill, Tilden Tool Mfg. Co., 1995 N. Fair Oaks, Pasadena, Calif., is offering a chart for determining the correct size concrete drill bit to use for any given size of pipe, conduit, machine bolt, or concrete anchor. It may be used as a wall chart or carried on jobs.

No. 7475: Cooling Tower Maintenance—A 16 page booklet containing hints on operating and maintaining water cooling towers has been prepared by C. H. Wheeler Mfg. Co., Lehigh and Sedgley Aves., Philadelphia 32. Information is given on putting the tower into service, inspection, lubrication, checking efficiency, and partial or total shutdown.

No. 7476: Corrosion Resisting Alloys and Equipment—General Catalog M of Duriron Co., Inc., Dayton 1, Ohio, is a 12 page condensation of data contained in 54 separate bulletins of the company. Various "Duriron," "Durimet," "Durichlor," and "Chlorimet" alloys are briefly described and data on composition, physical and mechanical properties, and corrosion resistance are given. Data are also given on numerous alloy products, including ejectors, fans, heaters, heat exchangers, steam jets, tanks, tank outlets, pumps, kettles, pipe and fittings, S bend condensers, nozzles, tower sections, and 16 different types of valves.

No. 7477: Corrosion Service Piping—Bulletin 485 of Taylor Forge & Pipe Works, P.O. Box 485, Chicago 90, is a comprehensive treatment on stainless steel and nickel alloy piping. Economics, standards, advantages of welding, extensive technical data, design tips, and complete

HIGH EFFICIENCY
More Horsepower Per Dollar
SHADED POLE MOTORS



5" dia. Frame

Type 470

6-pole, 1/8 to 1/15 H.P., maximum load speed 1050 r.p.m. furnished with single-speed, two or three-speed tapped windings, for fan or blower operation. Steel backed babbitt bearings. Ball-bearings, resilient mounting, conduit box, and thermal overload protection optional features. Send for Bulletin 4000.

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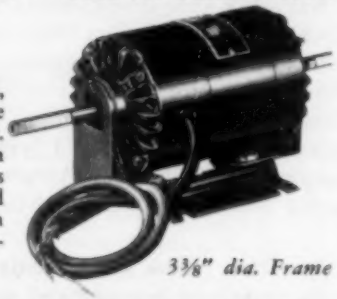

3 3/8" dia. Frame

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Division of Raytheon Mfg. Co.

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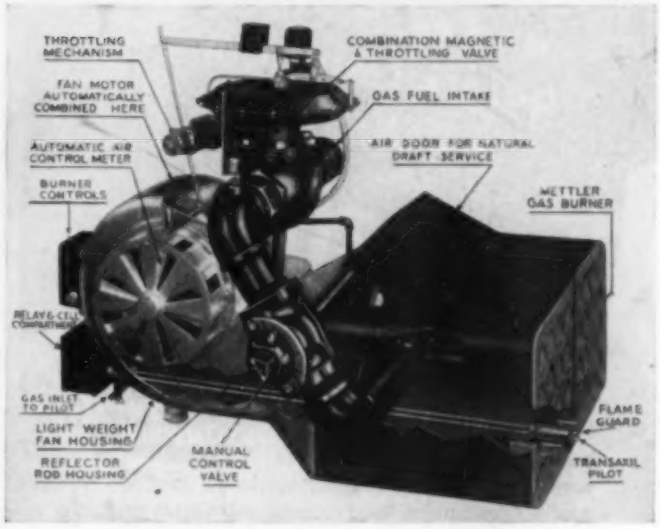
The Mettler "FAN-AIR" Gas Burner

LATEST MODEL FEATURES NEW AUTOMATIC SAFETY PILOT and TRIPLE CONTROLS



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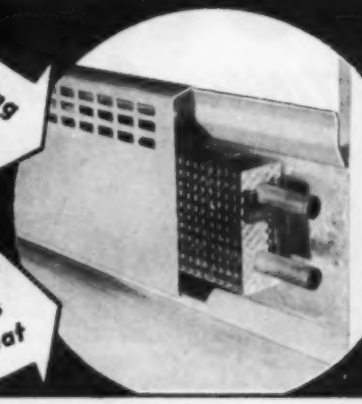

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- Smart looking, protrudes only 1 1/8".
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dimensional information on the company's stainless fittings and flanges are included in the 32 page booklet. The American Standard for Stainless Steel Pipe, ASA B36.19-49 has recently been adopted by the American Standards Association. This bulletin was prepared with the standard as its basis and it contains a discussion and explanation of the importance of the standard.

No. 7478: Duct Insulations—Design data for the application of "Fiberglas" duct insulating materials are contained in a new 12 page bulletin issued by Owens-Corning Fiberglas Corp., Toledo 1, Ohio. Properties of materials are described and tabulated, and a two page section provides a detailed description of field application procedure.

No. 7479: Electronic Relay Switches—Developed by Coral Designs, P.O. Box 248, Forest Hills, N. Y., units are available for use in automatic controls and gaging devices. They are said to provide means of using simple closure units as two position, three position, or floating type controllers for liquid level, pressure, flow, torque, position, humidity, temperature, etc. They are described in the Vol. 3, No. 2 issue of the company's house organ.

No. 7480: Flow Instruments and Control Valves—A wide variety of primary and secondary flow instruments and control valves for virtually all industries is described in a new 24 page illustrated bulletin released by Fischer & Porter Co., 50 County Line Rd., Hatboro, Pa. Covered are variable area, primary flow meters; electrical transmitters; flow rate recorders, controllers, and ratio controllers; totalizers; and direct connected recorders, controllers, and pneumatic transmitters. Also described is a new and improved "Pneumatrol" air operated control mechanism and the company's line of air operated diaphragm motor valves.

No. 7481: Guiding and Anchoring Expansion Joints—The Vol. 8, No. 2 issue of the house organ published by MagniLastic Div., Cook Electric Co., 2700 N. Southport Ave., Chicago 14, explains the need for correct guiding and anchoring of both packed and packless expansion joints. Faulty practices and their results are illustrated.

No. 7482: Heat Absorbing Glass—The story of how "Aklo" glass filters daylight to exclude solar heat and provides easier seeing is told in a new folder issued by Libbey-Owens-Ford Glass Co., Nicholas Bldg., Toledo 3. The glass is blue-green in color and is generally frosted.

No. 7483: Heaters, Tanks, Vent Valves, Other Products for the Chemical Industry—A new 20 page catalog, recently issued by Black, Sivalls & Bryson, Inc., 720 Delaware, Kansas City 6, Mo., describes gas dehydrators, tubular heaters, indirect heaters, chemical feeders, gas scrubbers, wood and steel tanks, vent valves, and vent valve-flame arrestor combinations. Charts give capacities, ratings, and specifications.

No. 7484: Heat Transfer Products—Catalog No. 148 of Young Radiator Co., Racine, Wis., is a 20 page booklet illustrating and briefly describing its line of heat transfer products. Equipment items include convectors, unit heat-

—But what PERFORMANCE

• The very smallness of the Skidmore Vertical Type "CV" Condensate Pump and Receiver, makes it ideal where space is limited. Yet, the Vertical Type "CV" Pump affords big PERFORMANCE with a wide range of SELECTIVITY. Adapted for condensate returns from all types steam heating and process equipment.

Capacities ranging from 500 to 10,000 sq. ft. E.D.R. Discharge pressures from 10 to 20 pounds. Write today for Bulletin No. 21, Dept. H.P.

Skidmore Pumps

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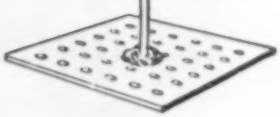

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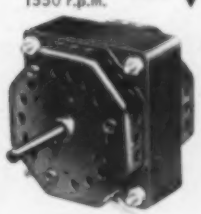
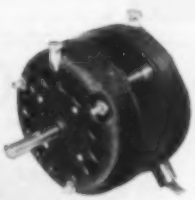
MODEL B
4-pole shaded pole motor. Approx. 1/30 h.p. 1550 r.p.m.



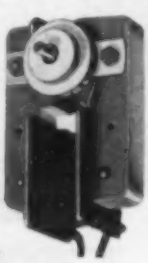


MODEL A
6-pole shaded pole motor. Approx. 1/30 h.p. 500 to 1050 r.p.m.

MODEL MS
2-pole shaded pole motor — full load h.p. .0021. Full load 2800 r.p.m.

Typical Fan Motor Uses

- Air Circulators
- Room Heaters
- Exhaust Fans
- Coolers
- Air Conditioners
- Unit Heaters
- Hair Dryers
- Controls



Write for catalog and specifications

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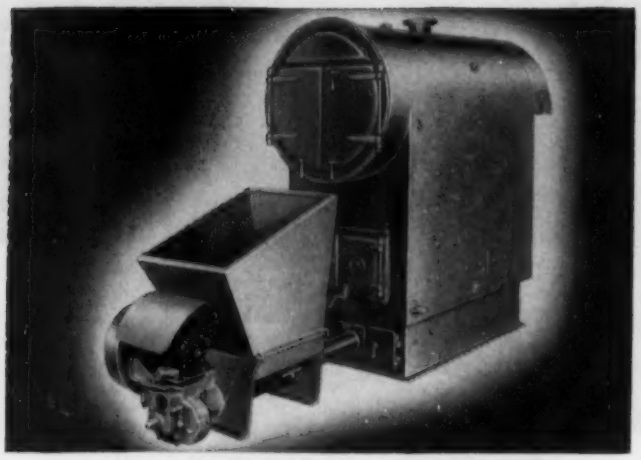
Export Department: 401 Broadway, New York 13, N. Y., U. S. A.



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For complete information, ask for Bulletin SF-1. Do it now while you're thinking of it.

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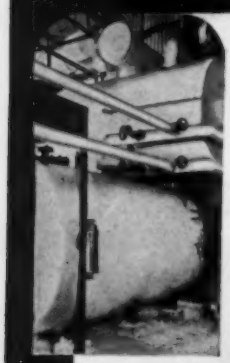

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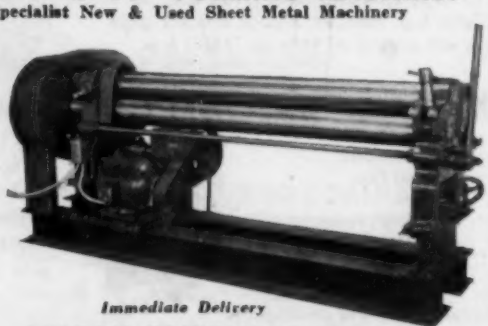
Specialist New & Used Sheet Metal Machinery

SIZES 1" to 6" Dia. Rolls 16" to 10' Long

ROLLS UP TO ¼" PLATE



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404-408 RACE ST. PHILADELPHIA 6, PENNA.

ers, radiators, air conditioning units, finned heat transfer coils, tube and bundle heat exchangers, jacket water coolers, cooling and condensing units, and specialized products.

No. 7485: Industrial and General Purpose Relays—Catalog D-20A of Ward Leonard Electric Co., Electronic Distributor Div., 53 W. Jackson Blvd., Chicago 4, lists industrial and general purpose relays which are carried in stock for immediate shipment. In addition to illustrating and describing the various types, the booklet gives contact ratings, coil specifications, sizes, and list prices.

No. 7486: Metallizing Packing Areas—The Vol. 4, No. 10 issue of the house organ published by Metallizing Engineering Co., Inc., 38-14 30th St., Long Island City 1, N. Y., contains a number of case histories which describe the advantages of metallizing packing areas. Rebuilding jobs mentioned include a shaft on a 5000 hp compressor, forging press rams, and pump sleeves.

No. 7487: Product Drying—Bulletin SGD 5 is the latest addition to the air dryer Datagram series of Bryant Industrial Div., Affiliated Gas Equipment, Inc., 1020 London Rd., Cleveland 10. Dealing with the application of the company's silica gel air dryers for product drying, it has been prepared to aid in the proper selection of equipment. The pamphlet is not a ready reference piece but may be used for guidance in solving an actual problem.

No. 7488: Pumps, Radiation, Unit Heaters, Heating Specialties—C. A. Dunham Co., 400 W. Madison St., Chicago 6, has prepared a two color wall chart, 22 in. x 32 in., showing its line of heating products. Charts are available to contractors for use on walls or for window display.

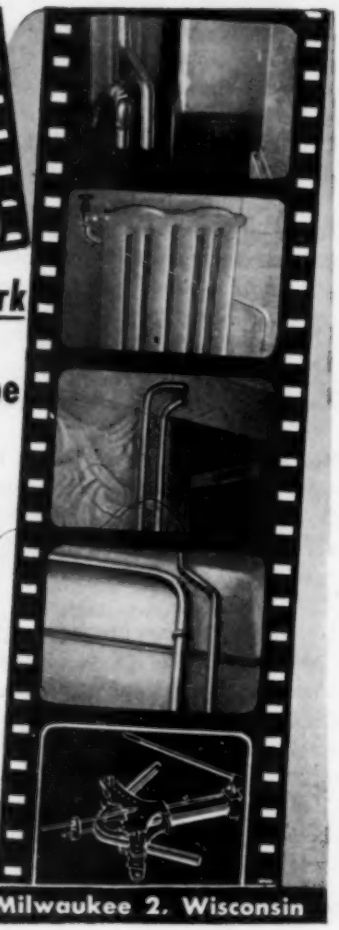
No. 7489: Radiant Heating—A new handbook has been issued by the Copper & Brass Research Association, 420 Lexington Ave., New York 17. Prepared primarily for heating engineers and architects under the direction of J. M. van Nieukerken, engineering consultant, this brochure presents a simple method of designing low temperature radiant panel heating systems using copper tube. The method is intended for use in connection with buildings of such size that "extremely refined" design is not justified.

No. 7490: Refrigeration Heat Exchanger, Low Temperature Defrost Unit—Economic advantages, ratings, sizes, weights, and prices of its liquid-to-suction heat exchangers are given in a leaflet issued by Refrigeration Economics Co., Inc., Canton, Ohio. Another leaflet covers its line of low temperature automatic defrost units. Each unit consists of two fans and two evaporators. One coil defrosts while the other continues to refrigerate and to supply the hot gas required to defrost.

No. 7491: Research and Development Laboratories—The expanded facilities of the Franklin Institute Laboratories for Research and Development, Benjamin Franklin Parkway at 20th St., Philadelphia 3, are described and illustrated in a recently published brochure. The organization is devoted primarily to development and to associated fundamental research.

SEE— for yourself

How a fast-running repair job is done in an old hotel



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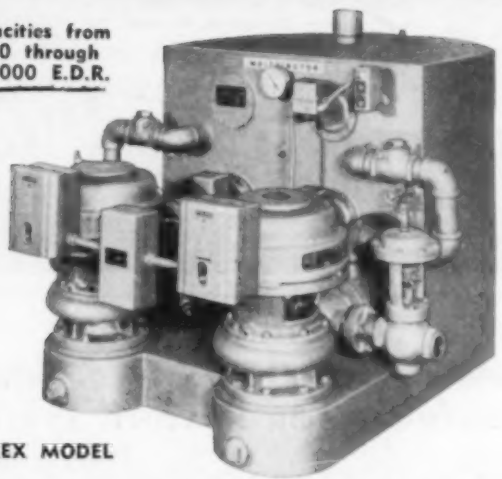
TAL Portable Bender

Work goes faster when you bend on the job — either original installation, radiant heat or repair. Fewer joints to leak in the future, less friction, more satisfaction for your customers and a bigger profit for you. Do as hundreds everywhere are already doing. Write for illustrated data bulletin. Dept. 13

Tal Bender, Inc. Milwaukee 2, Wisconsin

Vacuum Heating Pumps

Capacities from 2,500 through 300,000 E.D.R.



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Whittington Vacuum Heating Pumps maintain 28" Hg vacuum at shut-off conditions, using 70° water; thus can handle water nearer the evaporation point. Operate on economical water-jet principle, offering greater capacity at normal vacuum range with no loss of capacity during life of pump. Nationwide sales and engineering service. Write for Bulletin HP-49. Whittington Pump & Engineering Corp., 1124 Prospect St., Indianapolis 3, Indiana.

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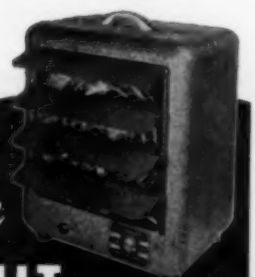
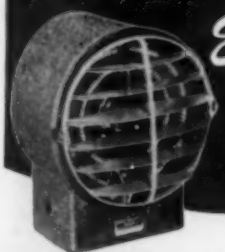
FOR COLD MORNINGS, HARD-TO-HEAT PLACES



CHROMALOX Electric UNIT HEATERS



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No. 7492: Self Operated Regulator—Bulletin 32[illegible] of Powers Regulator Co., 2720 Greenview Ave., Chicago 14, gives information on the No. 11 self powered regulator, used on water and oil heaters, crude oil treaters, tanks, vats, kettles, dryers, etc. Tables for selecting the proper size of regulator are included and applications are described.

No. 7493: Solenoid Valves—Construction features, dimensions, electrical data, and other pertinent information on solenoid valves of Skinner Electric Valve Div., Skinner Chuck Co., Norwalk, Conn., are given in three new bulletins. Stainless steel units, two-way normally closed, and three-way solenoid valves are covered in Bulletins 491, 494, and 493 respectively.

No. 7494: Suction and Water Hose Couplings—Combination male hose nipples, brass and malleable suction and water hose couplings, hose pipe, and adaptors are described and illustrated in Bulletin No. 5 of Hose Accessories Co., Lehigh Ave. and 17th St., Philadelphia 32.

No. 7495: Testing Mechanical Draft Cooling Towers—A series of quiz sheets, issued by Marley Co., Inc., Kansas City 15, Kan., provides general information on cooling towers and highlights features of the company's line. No. 8 in the series is devoted to testing mechanical draft towers and deals with such factors as instruments to be used, how and where should water and air temperatures be taken, what conditions should prevail during the test, and how should the test be conducted.

Statement of Ownership and Management of

HEATING, PIPING & AIR CONDITIONING

for October 1, 1949

The following is a statement of ownership, management, etc., as required by the act of Congress of August 24, 1912, as amended by the acts of March 3, 1933 and July 2, 1946 (Title 39, United States Code, Section 233) of Heating, Piping & Air Conditioning, published monthly at Chicago, Ill., for October 1, 1949:

1. The names and addresses of the publisher, editor, managing editor, and business manager are:

Publisher, F. P. Keeney, Chicago, Illinois.

Editor, C. M. Burnam, Jr., Chicago, Illinois.

Managing Editor, C. M. Burnam, Jr., Chicago, Illinois.

Business Manager, Chas. E. Price, Chicago, Illinois.

2. The owner is: (If owned by a corporation, its name and address must be stated and also immediately thereunder the names and addresses of stockholders owning or holding 1 percent or more of total amount of stock. If not owned by a corporation, the names and addresses of the individual owners must be given. If owned by a partnership or other unincorporated firm, its name and address, as well as that of each individual member, must be given.)

Keeney Publishing Company, 6 North Michigan Avenue, Chicago 2, Illinois. Stockholders: F. P. Keeney, Chicago, Illinois; W. I. Osborn, Fairfield, Conn.; Chas. E. Price, Chicago, Illinois; Robert A. Jack, Cleveland Heights, Ohio.

3. The known bondholders, mortgagees, and other security holders owning or holding 1 percent or more of total amount of bonds, mortgages, or other securities are: None.

4. Paragraphs 2 and 3 include, in cases where the stockholder or security holder appears upon the books of the company as trustee or in any other fiduciary relation, the name of the person or corporation for whom such trustee is acting; also the statements in the two paragraphs show the affiant's full knowledge and belief as to the circumstances and conditions under which stockholders and security holders who do not appear upon the books of the company as trustees, hold stock and securities in a capacity other than that of a bona fide owner.

F. P. Keeney, Publisher.

Sworn to and subscribed before me this 16th day of September, 1949.

Grace E. Waymire.

(SEAL) [My commission expires February 10, 1950]

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